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March, 1940

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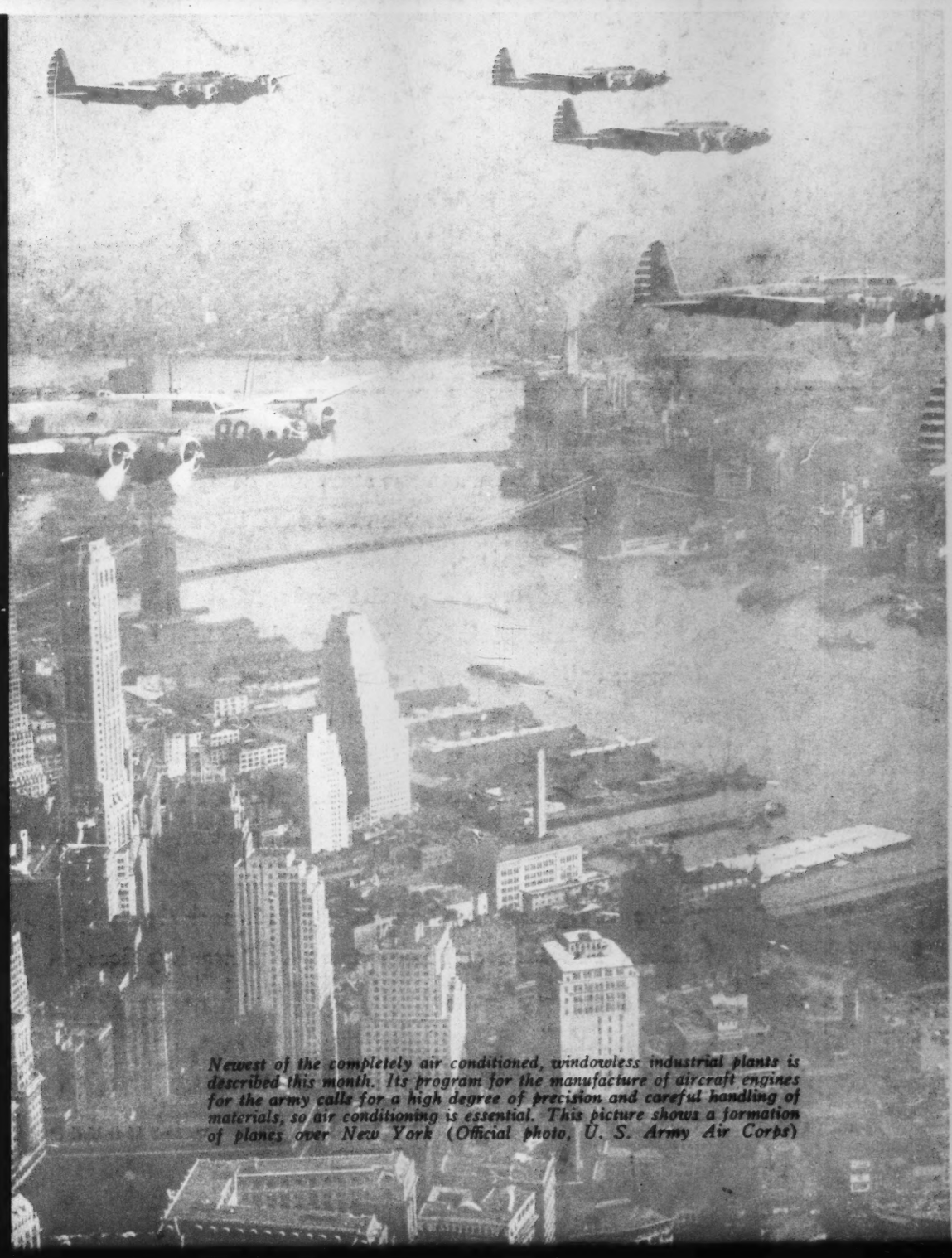

Newest of the completely air conditioned, windowless industrial plants is described this month. Its program for the manufacture of aircraft engines for the army calls for a high degree of precision and careful handling of materials, so air conditioning is essential. This picture shows a formation of planes over New York (Official photo, U. S. Army Air Corps)

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Journal Section
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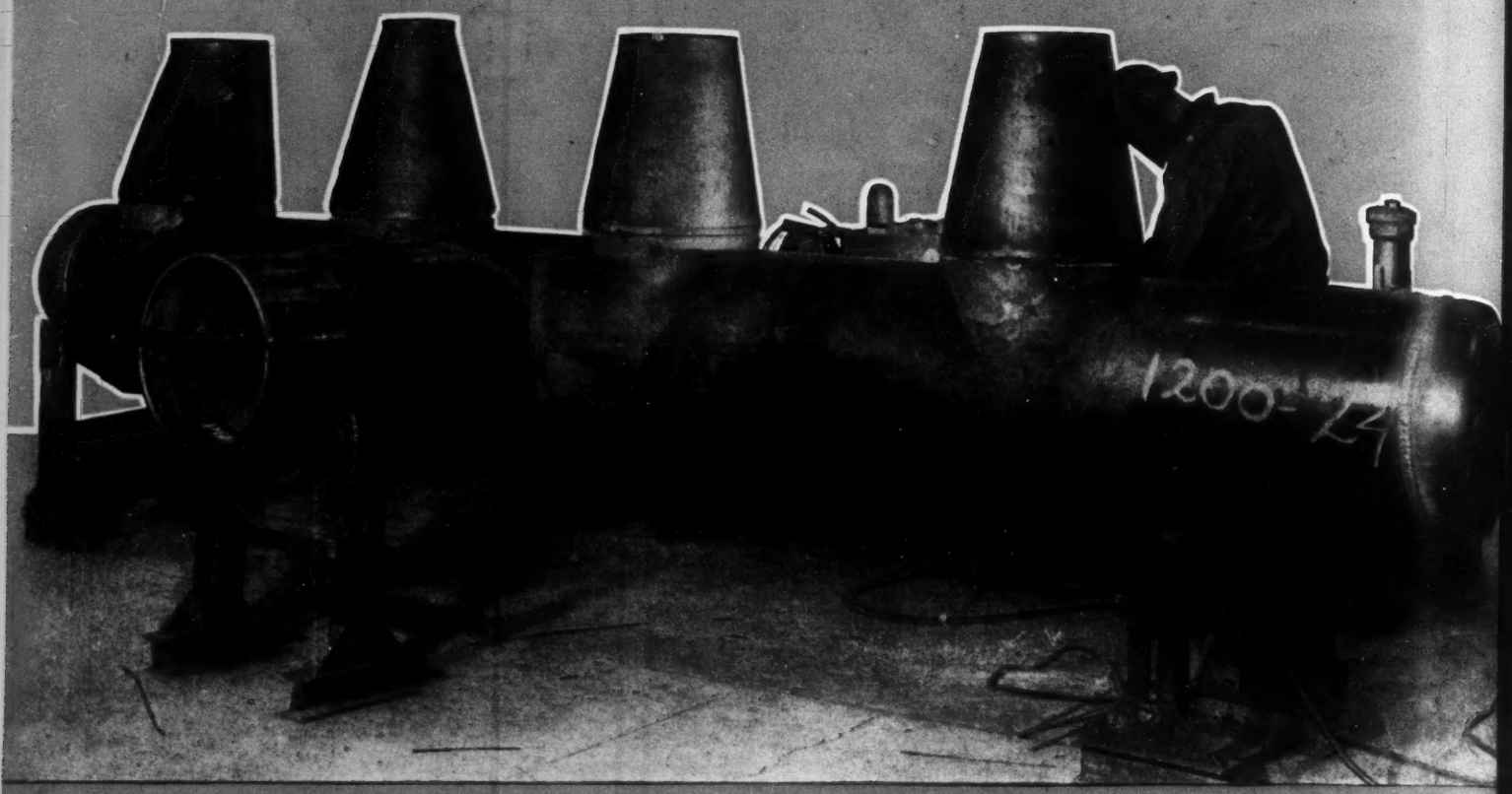
JOBS LIKE THIS SAVE WEEKS

between Investment and Return



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On jobs calling for special piping, Grinnell Prefabrication cuts days or weeks from this costly interval. While the plans are still on the boards, call in Grinnell. On schedule, accurately prefabricated piping sub-assemblies are delivered to the job. All but the simplest field welding has been performed with Grinnell's advanced prefabricating equipment.

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Design, Installation
Operation, Maintenance
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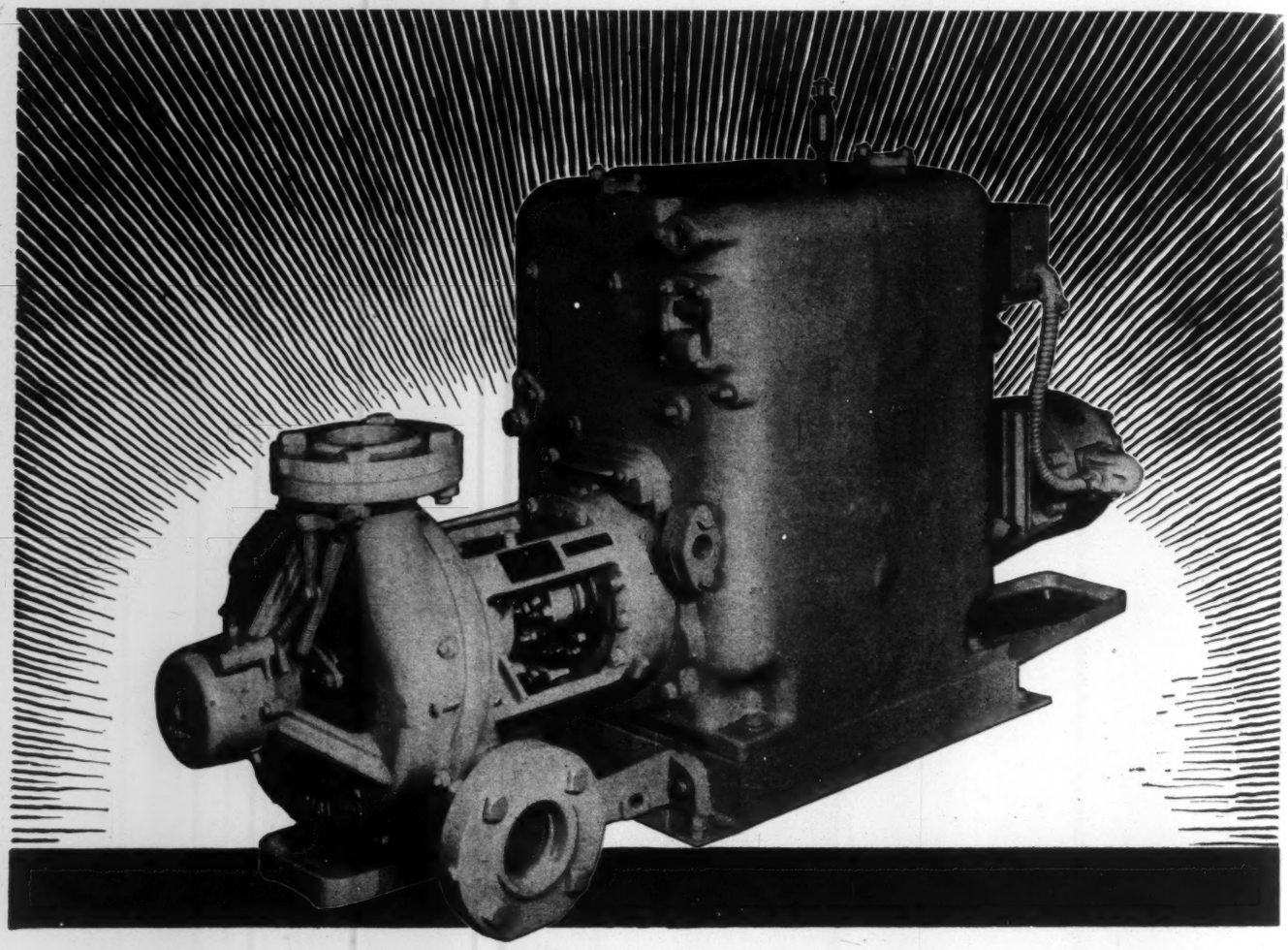
Safety!

THE NASH VAPOR TURBINE HEATING PUMP IS NOT STOPPED BY ELECTRIC CURRENT FAILURE



For service in greenhouses, hospitals, schools, theatres, and public buildings, where heat failure is a serious matter, the Nash Vapor Turbine Return Line Vacuum Heating pump should be specified, for this pump operates perfectly without electric current.

This is because the motive power is a special turbine operating on steam direct from the heating system. Steam used to drive the Vapor Turbine is returned to the system for heating, with little heat loss.

This pump insures constant heat in the system, it eliminates electric current cost, and it promotes high efficiency in the heating system. Heating efficiency is due to the fact that this pump can operate continuously with economy. Continuous operation means uniform circulation, and uniform conditions in the system mean steam saving.

This pump has but one moving part, no internal wearing parts, and requires no internal lubrication. Bulletin D-246 tells all about it, and it is free on request.

THE NASH ENGINEERING COMPANY

SOUTH NORWALK, CONNECTICUT, U. S. A.

THE EDITOR'S PAGES

BELIEVED IN HOTEL AIR CONDITIONING

The late Ralph Hitz, president of the National Hotel Management Co. which controls eight leading hotels in the United States, was a firm believer in hotel air conditioning. His career in the hotel business was largely marked with engineering tie-ups such as air conditioning, and the question of isolated generation of power.

Shortly after he was made manager of the Gibson Hotel in Cincinnati, he invested a quarter of a million dollars in remodeling and air conditioning. Later on in New York, the Hotel Lexington was modernized through his efforts, the program including air conditioning for what is now the Hawaiian room and which was previously a dining room above the laundry and ordinarily so hot that it could be used only in winter. When the Hotel Adolphus in Dallas had a particularly good year due to the centennial exposition there, plans and specifications were worked up and air conditioning was installed in 250 guest rooms. This made the hotel an outstanding example of comfort in the Southwest.

When Mr. Hitz's company took over the Belmont Plaza in New York, it was remodeled, the main dining room (creating the present Glass Hat) and the remodeled coffee shop both being air conditioned for year around comfort. The Hotel New Yorker always had air conditioning for the public rooms—except the lobby—and many thousands of dollars were spent in improving the air conditioning under Mr. Hitz's direction.

(The editors acknowledge with thanks the assistance of Warren D. Lewis, consulting engineer, National Hotel Management Co., Inc., in furnishing information on Mr. Hitz's interest in air conditioning.)

NEW SALES POLICY FOR GENERAL ELECTRIC

General Electric Co. has announced that central plant air conditioning and industrial refrigeration equipment of more than 5 hp will henceforth be sold directly to authorized contractors who, together with engineers and architects, will assume all application engineering and installation functions. Stuart M. Crocker, manager of the air conditioning and commercial refrigeration department, Bloomfield, N. J., has appointed Samuel Martin, Jr. head of the new contractor sales division. Products for sale to contractors include the full line of larger condensing units, ranging approximately from 5 to 60 tons in capacity; all evaporative condensers that fall within the 5 to 50 ton classification; and the central plant conditioners, of which the largest sizes are capable of more than 65 tons cooling, or 1½ million Btu per hr heating.

"By our changes we are recognizing the inherent differences between packaged products and engineering equipment," stated Mr. Crocker when explaining the new contractor selling set-up. "The engineering and installation of central plant air conditioning and industrial refrigeration jobs requires specialized facilities, which in many cases cannot profitably be maintained by distributors of packaged goods. We feel that today maintenance of such facilities logically falls within the province of engineers, architects and contractors who specialize in such work. Therefore, as regards central plant air conditioning and industrial refrigeration equipment, GE will concentrate on manufacturing and no longer will compete with specializing agencies in matters of application engineering and installation—nor will GE distributors be encouraged to do so."

FAN MAKERS REPORT PROGRESS

Progress in the Propeller Fan Manufacturers Association's nationwide campaign advocating certified ratings for propeller fans was reported at the association's recent annual meeting in Cleveland. Keynote of the meeting was that the purchaser should insist on certified ratings, that air delivery on any propeller fan equipment he purchases shall be in accordance with the standard test code for centrifugal and axial fans.

A. R. Stephan, DeBothezat Ventilating Equipment Co., was elected president, and M. W. Bauer, Aerovent Fan Co., was named vice-president. V. C. Shetler is secretary-treasurer.

BUILDING MODERNIZATION DURING 1940

According to an annual survey made by *Buildings and Building Management* reported in the January 16 issue of that publication, the percentages of building owners and managers planning to use the following products for modernization of office and commercial buildings during 1940 are: Air conditioning, 18 per cent; stokers, 6 per cent; temperature control, 9 per cent; oil burners, 5 per cent; boilers, 9 per cent; water heaters, 7 per cent; radiators, 10 per cent; pumps, 7 per cent; radiator traps and valves, 19 per cent; and piping, 13 per cent.

The corresponding figures for apartment building modernization are: Air conditioning, 6 per cent; water heaters, 26 per cent; temperature control, 14 per cent; stokers, 19 per cent; boilers, 19 per cent; oil burners, 15 per cent; radiators, 12 per cent; pumps, 9 per cent; radiator traps and valves, 26 per cent; and piping, 22 per cent.

INDUSTRIAL AWARD PROGRAM EXPLAINED

A 48 page brochure on its $200,000 industrial progress award program has been published by the James F. Lincoln Arc Welding Foundation. Official document of the 2½ year industrial progress program announced by the foundation in January, the booklet reflects the experience in conducting the previous award program in 1937-38 and incorporates comments and suggestions of many authorities.

The brochure announces the purpose of the program, indicates its wide scope and lists the 12 classifications and 46 divisions for participation which cover "every conceivable branch of the industrial field." Requests for copies should be addressed to the Secretary, James F. Lincoln Arc Welding Foundation, Cleveland, Ohio.

AMERICA'S JOB AT HOME, SAYS DU PONT

"Although certain industries may profit temporarily from war business, you and I know the aftermath of war will be a headache," Henry B. du Pont, vice-president and member of the executive committee of E. I. du Pont de Nemours & Co. told the Michigan Manufacturers' Association, which met recently in Detroit. American industry's job is at home, where "technical developments of the past few years have laid the foundation for additional millions of jobs," said Mr. du Pont.

"Estimates made by the National Industrial Conference Board and others indicate," he said, "that, if production of durable goods and construction had continued during the past nine years at the same rate as during the preceding nine years, there would have been

around 100 billion dollars more output than was actually realized.

"To make up only half the accumulated deficiencies would require the labor of several million additional men for many years," he continued. "Merely to regain and maintain the old standard of living reached during the 1920's, as measured by per capita production and consumption, would necessitate the operation of our factories and mines during the next 10 years at an average rate of activity far higher than we have ever known. There would be jobs for all able-bodied workers, and in boom periods there would be a labor shortage."

INDUSTRIAL CONDITIONING JOBS SHOW INCREASE

A feature of 1939 air conditioning business was the marked increase in orders from industrial concerns during the closing months of the year and the use of air conditioning in the manufacture of new products, according to Carrier Corp.'s annual letter to its stockholders.

A net profit in 1939 of $69,099.61 after all charges including inventory adjustments, taxes and depreciation was announced for the corporation. This profit compares with a loss of $1,133,021.42 in 1938. Orders booked during the entire year were approximately 12 per cent greater than in 1938 and completed sales for 1939 were 8 per cent above those of 1938. Unfilled orders at the close of the year were 32 per cent higher than in the previous year.

Employment of more than 200 new men by Carrier since January 1 was hailed last month by J. I. Lyle, president, as "not only a hopeful sign for general business, but also as an indication that air conditioning demand for large scale industrial installations was maintaining the pace set last year." The increase in business is expected to result in a 20 per cent jump in sales in 1940 over 1939, he said.

STOKER SALES BREAK RECORD

Factory sales of mechanical coal stokers in 1939 broke all records, according to figures released by the U. S. Department of Commerce, the Committee of Ten reports. One hundred and one manufacturers reporting to the department in 1939 sold 104,289 stokers of all kinds and sizes compared with 96,433 units in 1938 and 101,808 machines in 1937; an increase of 8.1 per cent over 1938 and 2.4 per cent over 1937 (The latter year held the existing record until 1939.) One hundred and twelve establishments reported sales in 1938 and 108 in 1937. Eleven companies discontinued a stoker line in 1939, the bureau reports.

The manufacturers whose data are included in these statistics produced approximately 95 per cent of the total dollar volume of the output of the industry as reported in the census of manufacturers for 1937. It is estimated that, taking into account the sales of companies not reporting to the government, there were approximately 110,000 stokers sold in 1939 with a small unknown percentage shipped to foreign countries. Unknown also is the number of new stokers sold and installed in 1939 which replaced old or obsolete models.

DEMONSTRATION OF TEMPERATURE, HUMIDITY

A simple but effective way of indicating to the layman the relation between temperature and relative humidity was suggested by executives of the York Ice Machinery Corp. at its recent series of sales conferences, where its new air conditioning and refrigeration equipment was presented to distributors and dealers. The demonstration requires only the use of a wide rubber band.

The length of the rubber band represents temperature, and its width represents the relative humidity of the air. Stretching the band signifies increasing the air temperature; the width of course decreases, as would the relative humidity on an increase in air temperature.

Dorothy Streitz pushes the button to start the "doll house" air conditioner modeled after the system in the National Broadcasting Co.'s quarters in Radio City, Rockefeller Center, New York City. This operating model, described by NBC Engineer Robert Close as being "correct in every detail" for the huge system serving the studios and offices, was built by hand by David Hillegas, who plans to install the miniature air conditioning system in a doll house for permanent exhibit. The model is in operation complete with water sprays, motors, fans, outlets, filters, and other necessary air conditioning parts.

Stewart E. Lauer, York's new president, announced at the conferences that John R. Hertzler has been appointed general sales manager. A graduate of Lehigh University in 1927, Mr. Hertzler is well-known through the papers on air conditioning which he has presented before various engineering meetings.

HONOR MODERN INDUSTRIAL PIONEERS

The National Association of Manufacturers sponsored last month its "modern pioneers" celebrations to honor inventors and research workers on "modern industrial frontiers." Another object of the "modern pioneer" programs was to pay tribute to the American patent system on its 150th anniversary.

Among the "modern pioneers" who received recognition were Dr. Willis H. Carrier, chairman of the board of Carrier Corp., for his work in air conditioning; John M. Frank of Ilg Electric Ventilating Co. for his development of the steam unit heater; Lewis Martin Ellison, Ellison Draft Gage Co., for his work on draft gages; Walter Geist of Allis-Chalmers Mfg. Co., for his work on the multiple V-belt drive; Harry E. LaBour, LaBour Co., Inc., for research and development in centrifugal pumping; and Ira E. McCabe, Mercoid Corp., for his work on automatic controls as applied to oil burners, stokers, air conditioning and other services.

REPORT NEW AIR CONDITIONED HOTEL

It was reported in the *Chicago Tribune* last month that Chicago is to have a $5,000,000 Statler hotel on North Michigan Ave. containing 1500 rooms. Of particular interest was the statement that every room would be air conditioned, as would all the public rooms (including a large convention hall).

According to the newspaper report, it was at first thought the company might buy a Chicago hotel, but it was decided it would be better to build a new one to take full advantage of the latest construction developments such as air conditioning.

MOVES TO LARGER SPACE

Due to the growth of its business it was announced last month that Maid-O'-Mist, Inc., no longer finds its available space at 180 N. Wacker Drive, Chicago, adequate and is taking over 10,000 sq ft of additional space at 215 N. Aberdeen St. The new location also provides better shipping and manufacturing facilities.

Heating · Piping and Air Conditioning

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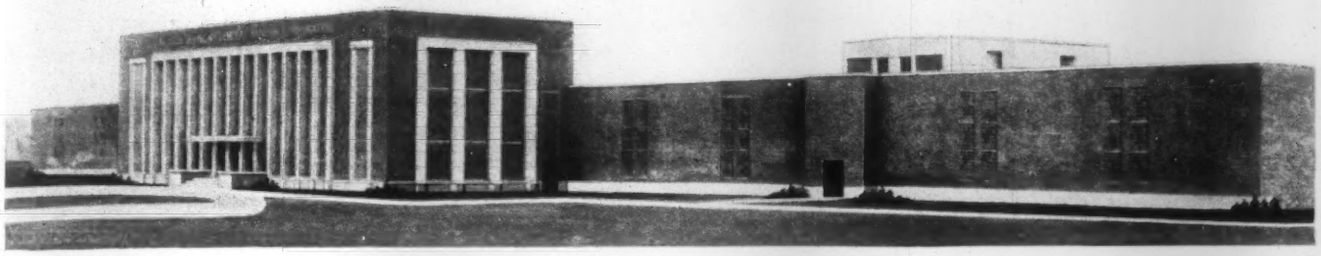


The new windowless, air conditioned plant of General Motors' Allison Division for the manufacture of army aircraft engines

Air Conditioning Vital to Precision in Manufacture of Aircraft Engines

THE world's newest version of a completely air conditioned, artificially lighted, windowless factory soon will be put into operation at Indianapolis, Ind., by the Allison Div. of General Motors Corp. The building and its equipment are unique in many respects; among its features are:

1) A combination of natural well water, "Freon" refrigerant, natural air and auxiliary oil burners in the air conditioning and heating scheme.

2) The first application of a new type of fluorescent lighting to a large scale industrial lighting problem, duplicating mid-day north light 24 hours a day.

3) Because of the unique combination of factors in heat insulation, air conditioning and artificial lighting it will take relatively less power to create this new approach to ideal working conditions.

4) Unusual paralleling of a diesel power plant with local utility electric lines to provide an adequate, uninterrupted source of power with unusual economy.

5) Unusual architectural effect, due to the windowless feature and use of glass brick panels.

6) An unusual tooling set up for precision work.

The new plant of the Allison Div. of General Motors Corp. at Indianapolis is the world's latest version of the air conditioned, artificially lighted, windowless industrial building. Its program for the manufacture of aircraft engines for the U. S. Army calls for a high degree of precision and careful handling of materials. Air conditioning is thus an essential in avoiding the effect of temperature changes on gages and precision machines, etching of highly polished metal surfaces because of perspiration, and for providing the comfortable conditions that are necessary because of the concentration and skill required of the workmen

"Manufacturing requirements in the new plant created conditions to be met which gave us an unusual opportunity to try out advanced thinking in factory construction," according to O. T. Kreusser, general manager of Allison Div. "Our program for the manufacture of aircraft engines for the U. S. Army calls for a high degree of precision and careful handling of materials. Temperature changes affect the micrometric gages and precision machines workmen constantly must use. Perspiration, in this climate, may cause etching on a highly polished metal surface, sometimes from just a sweaty finger print. Unusually fine lighting is required for accurate reading of gages and observance of work by operators. Comfort of workmen is unusually important because of the concentration and skill required. It was apparent when we laid out the plant that maintenance of a steady, comfortable temperature, proper humidity and dust count in the manufacturing building would be essential. Since air

conditioning and artificial lighting were required we were able to dispense with windows and monitor roofs, which contributed to economical construction."

General Plant Layout

Four new buildings on 53 acres of land comprise the plant; they include the manufacturing building with adjoining office building, a test building and a gate house. Welded construction of steel columns and trusses is used throughout.

A large receiving room, which is not air conditioned, at one side of the manufacturing building's main room, and other entrances serve as air locks so that unconditioned air does not rush into the air conditioned area when outside doors are opened. The office building, a three story structure which might be considered a front wing of the manufacturing building since the two are directly connected, is completely air conditioned, although it is not exclusively artificially lighted. The test building is a series of rooms with large open stacks. Propellers on engines on test pull the air in through one stack and push it out through another. The stacks are soundproofed. A small structure in which propellers used with the engines on test will be balanced adjoins the test building. The small gate house matches the rest of the plant architecturally.

Air Conditioning Is Vital

Because of the precision nature of many of the operations, controlled temperatures, humidity, dust count and lighting are vital in this plant.

A windowless building for the main plant was definitely determined upon, with the consequent necessity of complete air conditioning for all periods of operation. General Motors, in conjunction with the engineers of the Allison Div., therefore designed a system to combine the features essential to year 'round air conditioning for men at specific types of work. It was determined that 78 F effective temperature in the summertime is the proper temperature for men employed in occupations similar to those existing in this plant. However, the building, being windowless and requiring air conditioning at all times of the year and over a 24 hr work period, had to be designed to produce the maximum economies of operation and to be able to utilize to the maximum extent nature's assistance in the production of such conditions.

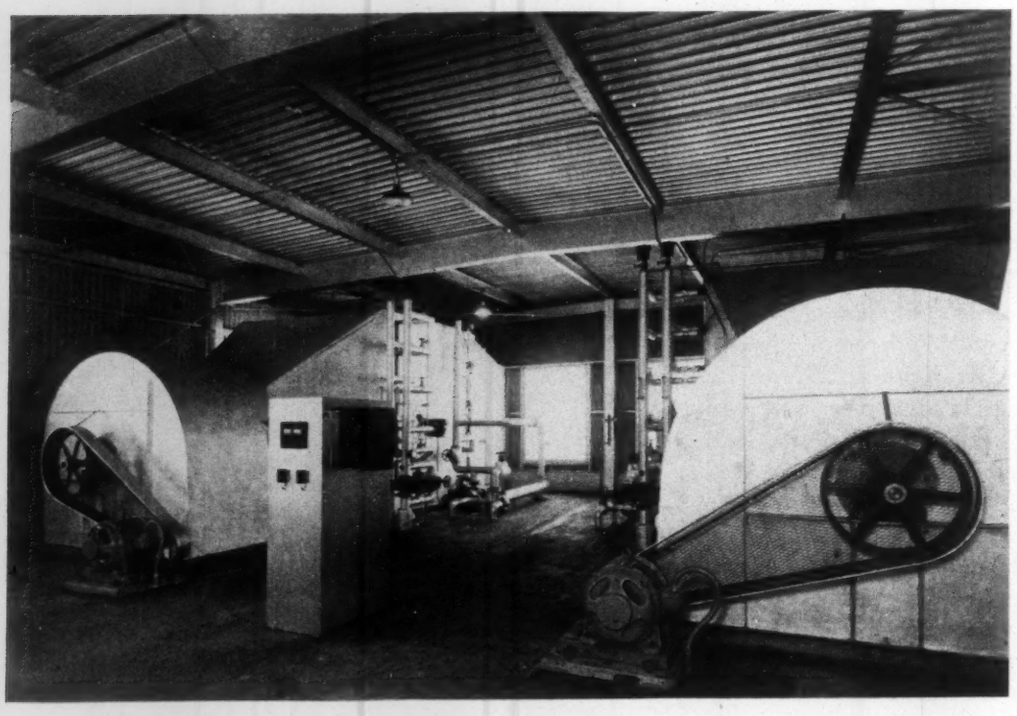
It was therefore required of this system that with outside conditions of 95 F dry bulb and 78 F wet bulb and a sun effect such as prevails in Indianapolis, that the maximum inside conditions, when operating at full capacity, should not exceed 85 F dry bulb and 72 F wet bulb (53½ per cent relative humidity). These conditions are equivalent to the 78 F effective temperature desired. The system was designed to maintain 70 F in wintertime with a 10 F outside temperature. The building was designed for an average of 1000 occupants, 1440 brake hp of motors, and 250 kw of lights.

It was calculated that 90,000 cfm of outside air would be required under minimum conditions to maintain an outward pressure and provide sufficient air for ventilation and the dilution of oil fumes which might arise from the machines. These factors created a cooling load under maximum conditions in the summer of 815 tons. As the conditions required in the building—that is, 85 F dry bulb and 72 F wet bulb—represent a saturation temperature of approximately 66 F, it meant that the air delivered into the factory under maximum conditions would have to be saturated at 66 F and that consequently an allowable rise of about 18½ F could occur to maintain the conditions specified. With this rise and a sensible heat gain of approximately 7,000,000 Btu per hr, some 356,000 cfm of air would have to be introduced to maintain the conditions.

To allow for the possibilities of breakdown or equipment servicing, and bearing in mind that the air conditioning system must function in order to make it possible for work to be carried on, it was decided to divide the system into eight similar, duplicate component parts.

To provide against the breakdown of any one of these systems, each was sized to handle 52,000 cfm. It was developed from an engineering angle that the amount of air required for summer cooling was more than sufficient for the winter requirements, so the system was designed on the summer basis.

One of the four air conditioning penthouses atop the plant. Eight separate central station conditioning units are used



Evaporative Cooling Used at Times

It was decided that if complete evaporation could be accomplished, that during a large proportion of the operating time evaporative cooling could be used to produce the necessary results, without requiring refrigeration. An air conditioner or washer of a type that would do both cleaning and saturation of the air at the wet bulb condition of the air entering it was therefore selected. These units are made up of cells packed with glass strands parallel to the air stream, over which water flows to give efficient heat transfer and cleaning.

Eight of these units, each capable of handling 52,000 cfm, were installed in penthouses on the roof of the building, situated at equally distant points. They were installed in pairs, four penthouses being erected in which to house the eight unitary systems. The units were designed so that with a change in the outside wet bulb more or less outside air could be taken into the building; in other words, as the wet bulb falls to a point below 66 F, all outside air will be introduced, or a mixture of outside and recirculated air will be introduced into the washers to maintain the required leaving dew point.

Substantial power saving is effected by the relatively small quantity of water required for saturation of the air and the low pressure required for the distribution of the water over the conditioner cells. Each cell is capable of handling from 1000 to 1100 cfm of air. For simple humidification and evaporative cooling 2½ gpm of water per cell distributed at 6 lb nozzle pressure is required. There are 48 of these cells in each of the conditioners or washers.

Air Handling and Distribution

The penthouses were so designed that they not only will act as mixing chambers for recirculated and outside air, but by automatically controlled louvers they also act as relief dampers, through which the excess air taken from inside can be relieved to the outside atmosphere.

The fans for delivering air to the eight sections are of the backwardly curved type, single width, single inlet, downblast discharge, discharging directly into the distributing system underneath. These fans are driven through belt drives by 30 hp motors. Due to the fact that the coils are used for both heating and cooling and consequently must be protected from direct moisture, two sets of eliminators are required with one set of coils. The one set of eliminators prevents entrained water from lodging on the coils when they are being used for heating purposes, which might occasion over-humidification. The second set of eliminators, which is used only in the summertime, is to prevent the condensate from the air deposited on the coils from being carried out to the system.

The fans each discharge into a practically identical duct system of the simplest type, but special diffusing and adjustable outlets permit a uniform distribution over the entire working area. As the system of supply is uniform, no return duct system is necessary, and the air returns to the penthouses by the creation of a pressure in the room, for either recirculation or for relief into the outside atmosphere. The fans, coils, conditioners, motors and pumps are mounted on concrete and cork foundations to eliminate condensation, dripping and vibration.

It was determined to cool the air by interchange, with coils through which cold water will be circulated when the wet bulb rises above the required point. The coils were especially designed so that in the winter they can be utilized for the circulation of hot water to increase the temperature of the delivered air in order to maintain approximately 70 F in the plant.

For producing the necessary cold water circulation, reciprocating "Freon" machines are employed. As this plant is to be operated by diesel engines, jacket cooling and condensing water are essential, and as the territory in which the plant was erected abounds in gravel strata well water around 54 F, it was decided to dig wells for the purpose of procuring water for air cooling, condensing purposes and for jacket cooling.

A heat balance investigation indicated that if the well water is first used for air conditioning purposes its temperature on leaving the air conditioning system will

The new type of fluorescent lighting used gives five times as much light with seven-tenths the power of older methods

still be amply low for condensing purposes and, after, for jacket cooling for both the diesel units and production test. However, it was found that the average water from wells in this territory is of such hardness that coils first subjected to cold water of this nature and then to heat were likely to deposit carbonates throughout the system. Therefore, this cooling water is used only indirectly; cold and hot water circulated in a closed system is used for heating and cooling, the same water being taken through an interchanger for cooling purposes or through a different interchanger for heating purposes, and bypassed either automatically or by hand, according to the season or the prevailing conditions. On investigation, it was discovered that 1350 gpm of 54 F well water is available, and that this will cool water (in an interchanger) coming back from the air conditioning system at 64 F, to 59½ F, the well water leaving the interchanger being warmed to a temperature of 60½ F—amply cool for condensing purposes and for jacket cooling. Moreover, it reduces the required refrigeration effect to be obtained from the "Freon" machines from 815 tons to 575 tons. The refrigeration machines are driven either by diesel or electric power, and are uniquely connected to the diesel and motor drives so that the diesel engines for driving the compressors can be used to drive the motors as generators to supplement the electric capacity of the plant.

The system of control is such that, automatically, the amount of outside air introduced into the system is varied from a minimum of 90,000 to a maximum of about 400,000 cfm, depending upon the outside wet bulb. The amount of refrigerated water is automatically regulated to produce 66 F saturated air leaving the coils. The return air dampers are regulated in step with the outside air dampers, so that when the outside conditions permit the introduction of large quantities of outside air, recirculation into the system is prevented and the air is relieved through the automatic louvers in the penthouses.

Air Conditioning and Heating Other Buildings

Office Building—The adjoining buildings were considered in the analysis of the entire project. One of these is the office building, which is to be conditioned to a lower effective temperature than that maintained in the main factory section. This system, which requires 21,000 cfm of air and is to be maintained at a 73 F effective temperature, for the time being is taking its air from the main system, hooking into the lateral distributing ducts of the main factory. The system is zoned for heating and has separate heaters in its own distributing system. The air from the office building is returned by fans into the main factory section.

Paint Shop—The only other section of the building which requires air conditioning is the paint shop, and this section, isolated from the main factory, has its own system.

Other adjoining buildings, not requiring air conditioning and not being of the windowless type, are not being conditioned for the time being, but are being heated and ventilated by unit heaters.

Forging and Casting Cleaning—For the promotion of greater workman comfort and health and better precision work, the forging and casting cleaning department in the main plant is equipped with a new arrangement for drawing off metal dust. The workmen sit at a long metal table upon which the machinery is placed. A slot is situated along the front edge of this table. A central blower sucks all small metal particles across the table surface down into tubes to a central exhaust pipe from which the dust is expelled at a point where it can do no harm.

The arrangement is completed by an ingenious idea for preventing dust from reaching the breathing area. The conditioned air enters the room at the top. Since the slots on the tables are directly below the workman's face and at about the level of his waist, the suction through the slot not only takes the metal particles off the table and from around the grinding machines, but also draws fresh, clean air down across the face of the workman. It is next to impossible for dust to get to his nose.

Possible Savings With Welded Construction

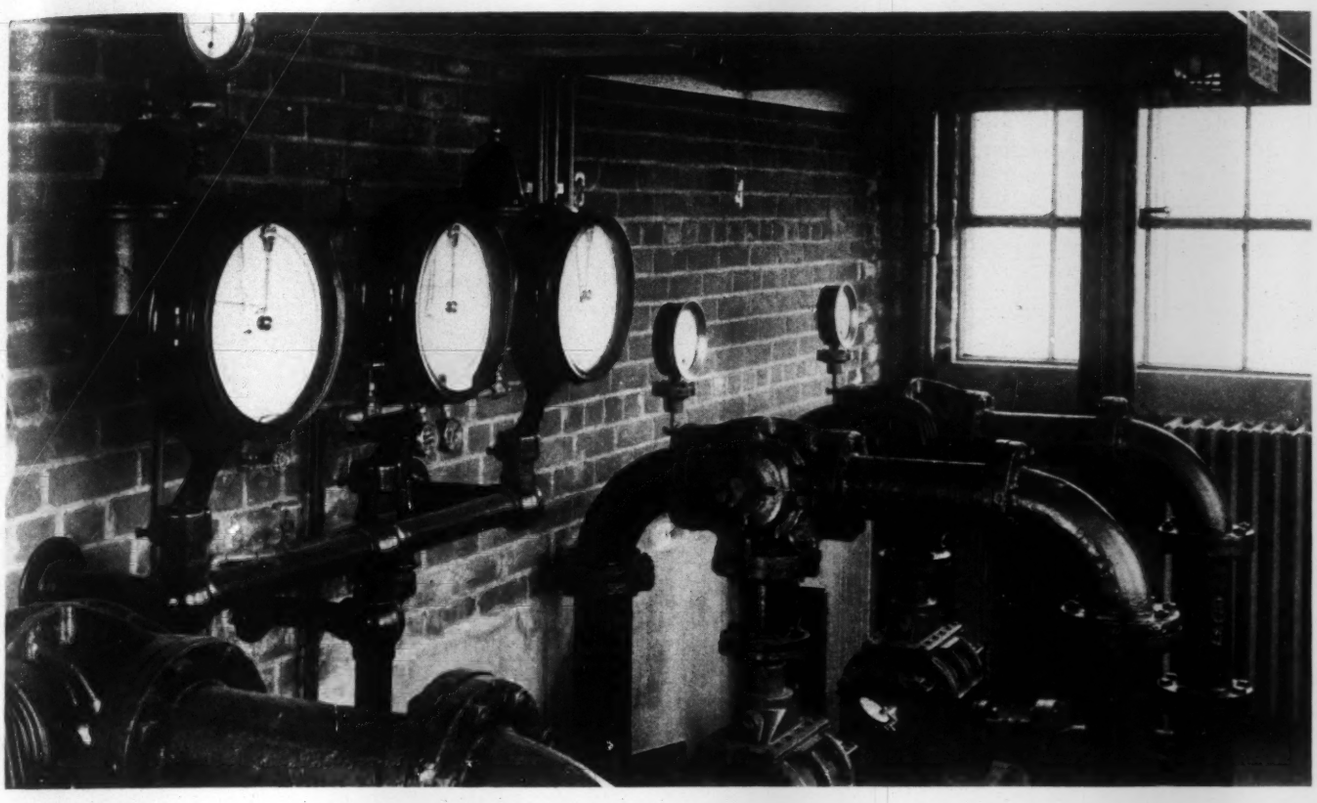
The papers which received awards in the "containers—contents moving" or piping classification in the first $200,000 award program of the James F. Lincoln Arc Welding Foundation were reviewed in the December, 1938, HEATING, PIPING AND AIR CONDITIONING, pp. 775-776 and 781. Savings of $1,600,000,000 were shown "available in industry by wider application of arc welding" in the various papers submitted for awards. Since publication of this earlier review, copies of some of the papers submitted which did not receive awards have been made available by the Foundation and a brief description of them is presented here.

In a paper on Improved Design of Refrigerant Control Piping for Fabrication by Arc Welding, submitted by Louis H. Roller, the author stated that arc welding piping found early application in the refrigeration industry because of its many advantages, among which are (1) refrigerant tight joints with strength equal to that of the pipe, (2) ability to use odd angle curves, (3) elimination of fittings in which leaks may develop due to porosity of castings which sometimes contain impurities with which the refrigerants react chemically, (4) closer assembly of lines due to little space required for welded connections, (5) ease of support by welding directly to piping, (6) reduced cost of galvanizing by cutting down weight, and (7) time saved in making up multiple assemblies or headers.

The purpose of his paper was to present an improved design of arc welded piping as applied to the supply and control of refrigerant for the evaporators of refrigeration systems, particularly those using ammonia. The basic principle of the design submitted was the combining of the separate liquid and suction risers of the usual flooded evaporator into one double pipe riser especially suited to fabrication by arc welding. According to the author, this combination results in better operating efficiency, and the use of arc welding shows a saving in fabrication cost and a lower weight of control piping.

In a paper on Welded Valve Bodies, submitted by B. F. Lownsbery, the author described a method of manufacturing a valve body of drawn tube, assembled by means of arc welding and not differing from a standard

[*Concluded on p.* 167]



Flow Meter Accuracy Governed by the Piping Layout, Pressure Tap Location

A READER who was interested in my article in the November, 1939, HEATING, PIPING AND AIR CONDITIONING on measuring flow in piping and flow meter applications comments as follows:

The article [in November] does not bring out two pertinent facts regarding meter installations which possibly should have been included. One is the fact that a great many meter installations are made in lines less than 2 in. nominal size. In cases of this sort it is considered good practice to swage to 2 in. for the meter run in order to insure accurate measurement.

As a second point, it would have been desirable to have made brief note of the proper location of the orifice taps in the flanges or line as the case may be. In other words, should pressure taps be brought from the bottom horizontal center line or from the top of the line? This point is frequently overlooked and often not thoroughly understood by field engineers.

This reader's comments bring out two points which are extremely important to accurate flow measurement. Fig. 1 shows the variation of static pressure in the neighborhood of an orifice. These data are plotted from the results of the Edgewood Arsenal tests of the Bureau of Standards, and are of sufficient accuracy for use as a means of correcting for deviations in pressure tap locations. It will be noted that, except for very low ratios of orifice to pipe diameter, the pressure changes rapidly from a point approximately one-half pipe diameter downstream from the orifice. These changes are of sufficient magnitude to constitute a large percentage of the differential pressure between the upstream and downstream pressure taps.

Under ideal conditions, a pressure tap of ¼ in. diameter may be satisfactory, but under most commercial operating conditions, a pressure tap of less than ½ in. opening clogs, freezes, or cakes up quickly and is impractical. In 2 in. pipe this is equivalent to one-fourth of a pipe diameter; in 4 in. pipe, to one-eighth of a pipe

By **L. K. Spink***

The information given here supplements an earlier article by Mr. Spink (published in the November HEATING, PIPING AND AIR CONDITIONING) which explained the importance of the relationship between piping design and layout and flow metering accuracy. Accuracy of flow meters depends very definitely on how and where they are installed in the piping . . . . . The author gives this month supplementary data on location of the pressure taps in the pipe line, and excerpts from a committee report on meter runs

*Manager, Meter and Gage Div., The Foxboro Co.

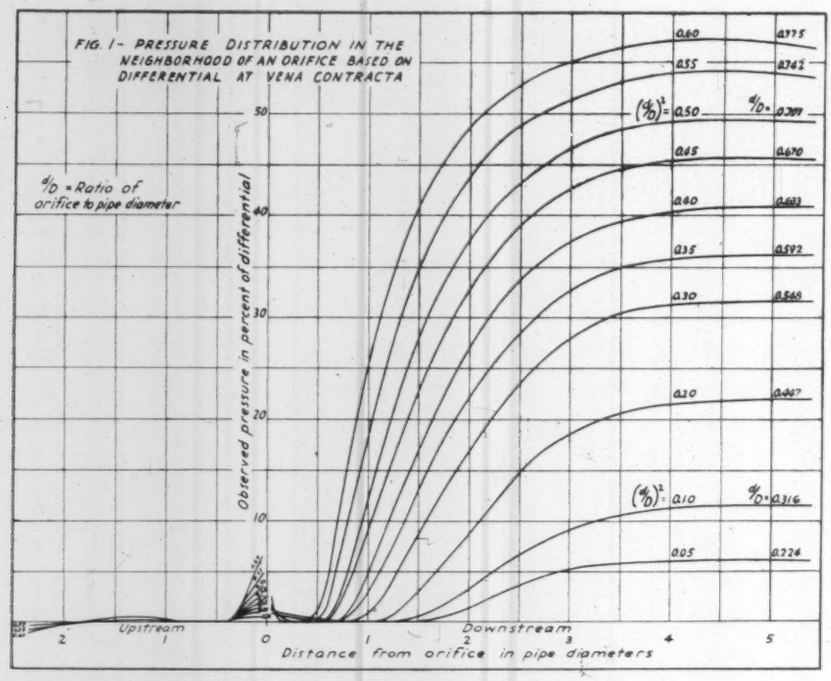


FIG. 1- PRESSURE DISTRIBUTION IN THE NEIGHBORHOOD OF AN ORIFICE BASED ON DIFFERENTIAL AT VENA CONTRACTA
d/D = Ratio of orifice to pipe diameter
Observed pressure in percent of differential
Upstream
Downstream
Distance from orifice in pipe diameters

diameter; in 6 in. pipe, to one-twelfth of a pipe diameter; in 8 in. pipe, to one-sixteenth of a pipe diameter; in 10 in. pipe, to one-twentieth of a pipe diameter, etc. Obviously, if a pressure tap is placed in this region of rapidly changing pressures, with the pressure at one side of the tap different from that at the other side of the tap, the resulting flow measurement is unreliable.

It is very desirable to keep the entire opening of the pressure taps in the steady pressure region; hence on the 2 in. line, the pressure taps should be centered not more than 1 in. from the face of the plate, and if ½ in. pressure taps are used, high ratios of orifice to pipe diameters should be avoided. On smaller pipes, rapid change in pressure makes impractical use of close-up taps of sufficient diameter to be practical for many commercial measurements. On pipe of 6 in. or less, the flat portion of the pressure curve downstream from the orifice usually comes within the limits of the flange hub, necessitating the use of tapped orifice flanges or orifice plate holding rings containing pressure taps. On larger lines the taps may usually be drilled into the pipe without interfering with the flange.

Pressure taps at two and one-half pipe diameters upstream and eight downstream may be used on the smaller pipe lines to avoid the region of rapidly changing pressure, but are more subject to pipe roughness and sacrifice something in accuracy. The American Gas Association recommendations give a 50 per cent greater accuracy tolerance for these connections and place closer tolerance on pipe diameter, besides lowering the range of usable ratios of orifice to pipe diameter.

Proper Axis for Pressure Taps

Regarding the question as to the proper axis for taking off pressure taps, this is also an important point. On gas, the best location is the vertical axis with the pressure taps running upward. This permits condensate to drain freely into the line of flow without clogging the meter leads and creating false heads. On liquids, the best location is the horizontal axis, which permits air or vapors to find their way out of the piping into the line of flow, and prevents air or vapor in the top of the line and sediment or water in the bottom of the line from carrying out into the instrument piping.

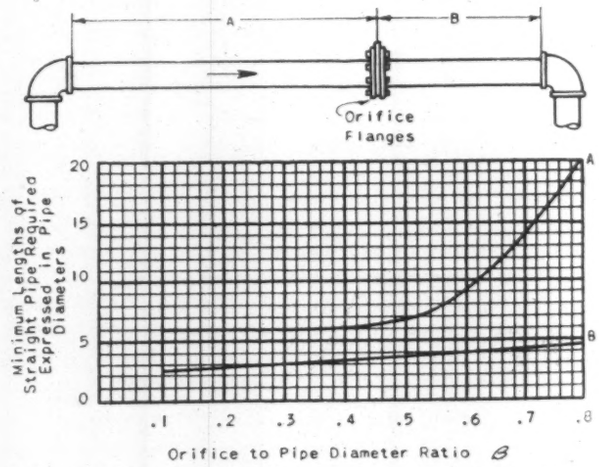
On steam, if the meter is below the line, the best location is on the horizontal axis with the condensing chambers on the same level. If the condensing chambers must be placed above the line, however, the taps should come out of the top of the line, which permits free counterflow of steam in an upward direction and condensate in a downward direction.

Condensing chambers are usually placed above the meter, which places them above the steam line if the meter is above the line. This is done because, in case of a shutdown, condensing steam pulls a vacuum on the system which expands any air or vapor in the system. If the condensing chambers are below the meter, this vacuum, added to that created by the siphon effect of the liquid between the condensing chambers and the meter, pulls the condensate out of the chambers and piping, with the result that the piping must be refilled after every shutdown.

Data from AGA Committee Report

The following data supplement the November article. The curves showing recommended straight runs of pipe for orifice meter settings represent the best judgment of a committee after studying test data gathered over a period of several years. They are an almost indispensable

I. Installation with One Ell or Tee Preceding the Orifice Run


A
B
Orifice Flanges
Minimum Lengths of Straight Pipe Required Expressed in Pipe Diameters
Orifice to Pipe Diameter Ratio d/D

Note 1. When "pipe taps"[1] are used, A should be increased by two pipe diameters and B by eight pipe diameters.

Note 2. When the diameter of orifice may require changing to meet different conditions, the lengths of straight pipe should be those required for the maximum orifice to pipe diameter ratio that may be used.

[1] "Pipe taps" as referred to in these recommendations are those located at 2½ pipe diameters upstream and 8 pipe diameters downstream from the orifice.

guide in laying out piping where the future installation of meters must be considered.

Although these recommendations were prepared primarily for gas measurement, they apply also to steam and liquids where the same high degree of accuracy is desired. With viscous liquids, it is probable that the disturbed flow damps itself out more quickly and therefore satisfactory measurement may be obtained with shorter runs of pipe. However, no curves, rules or formulas are available for taking this into account.

This material has been taken from the AGA gas measurement committee's report No. 2, with the permission of the AGA.

Meter Runs

Length of Pipe Preceding and Following an Orifice

The orifice flow constants and other factors used are based on the results of many experiments. In all cases, normal flow conditions were obtained by the use of long straight lengths of pipe, both upstream and downstream from the orifice, or by the use of straightening vanes. To obtain satisfactory measurement of flow, these conditions should be approximated in practice.

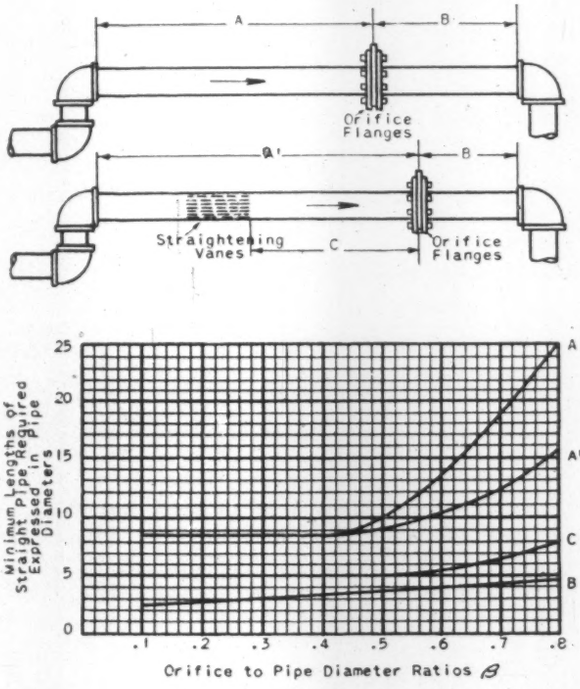
Any serious distortions of the flow will produce errors, and there are many locations at which the orifice will not produce satisfactory results. Certain of the more common types of installations have been studied as to their effect on metering accuracy, and the results of these studies are shown in the following sketches. These sketches show the minimum lengths of straight pipe required, both preceding and following an orifice, if the errors caused by such disturbances are to be eliminated or kept within the tolerance limits stated in this report.

The graph charts accompanying the installation sketches indicate that the minimum length of a straight pipe required varies with the orifice to pipe diameter ratio, longer lengths of straight pipe being required for the higher ratios. When the diameter of the orifice may require changing to meet different conditions, the lengths of straight pipe installed should be the same as that given for the *maximum* orifice to pipe diameter ratio that may be used. Meter runs longer than those indicated by the graphs are desirable when circumstances permit. When straightening vanes are used with longer meter runs the dimension indicated by the difference between A' and C should not be less than that indicated by the graphs.

The first installation sketch shows one ell preceding the straight run of pipe in front of the orifice, a very common type of installation. No tests have been run on this type of installation using a tee instead of an ell preceding the run of straight pipe; however, the same length of straight pipe will, it is believed, hold for an installation with a tee preceding the straight run as for an ell; further, that there will be no difference whether the run of the tee or the side of the tee joins the straight run preceding the orifice location.

The second installation sketch shows two ells or two bends preceding the straight run of pipe upstream of the orifice flange. There is more of a distortion of the flow lines in passing through

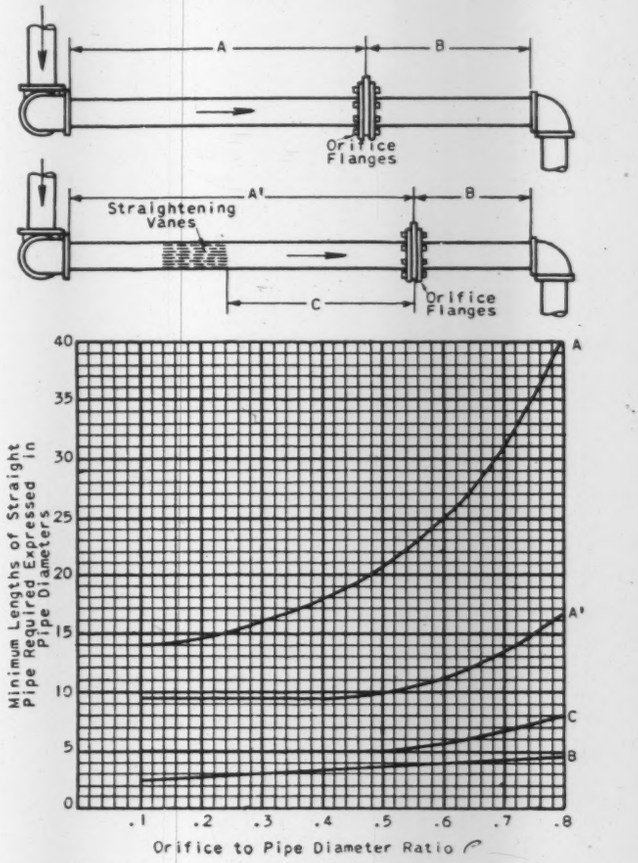
II. Installation with Two Ells or Bends Preceding the Orifice Run (Bends in Same Plane)



Note 1. When "pipe taps" are used, A, A' and C should be increased by two pipe diameters, and B by eight pipe diameters.

Note 2. When the diameter of the orifice may require changing to meet different conditions, the lengths of straight pipe should be those required for the maximum orifice to pipe diameter ratio that may be used.

III. Installation with Two Ells or Bends Preceding the Orifice Run. (Bends not in the same plane)



Note 1. When "pipe taps" are used, A, A' and C should be increased by two pipe diameters, and B by eight pipe diameters.

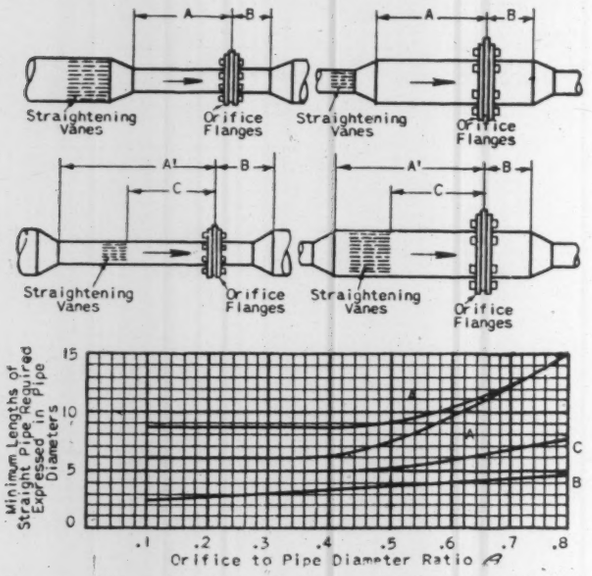
Note 2. When the two ells shown in the above sketch are closely preceded by a third which is not in the same plane as the middle or second ell, the piping requirements shown by A should be doubled.

Note 3. When the diameter of the orifice may require changing to meet different conditions, the lengths of straight pipe should be those required for the maximum orifice to pipe diameter ratio that may be used.

the two fittings than through one as shown in the preceding sketch, even though they are in the same plane.

The third installation sketch shows two ells, or two bends, at right angles to each other (not in the same plane). This type of bend produces considerable disturbance that is ironed out only by long runs of straight pipe, or by straightening vanes, or both on the inlet side of the orifice flange.

IV. Installation with Reducer Preceding the Orifice Run



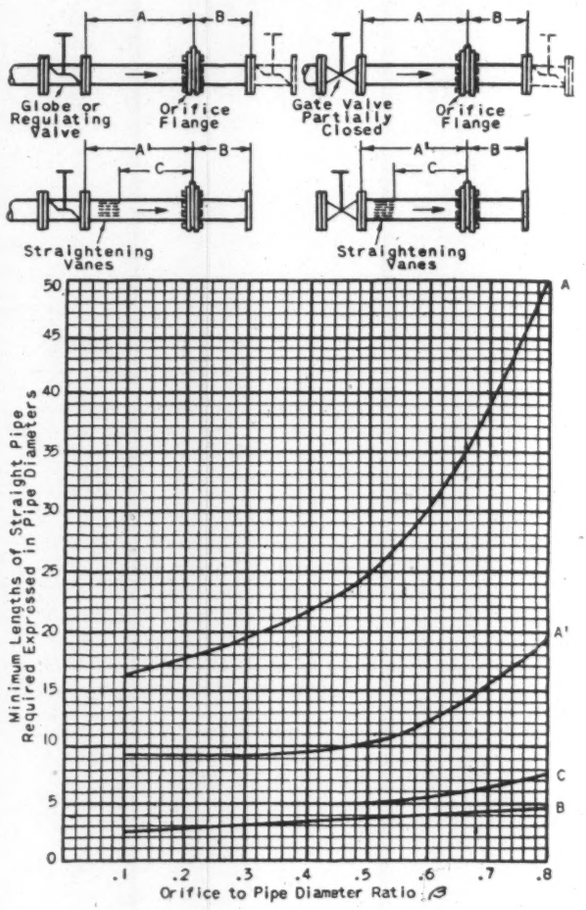
Note 1. When "pipe taps" are used, length A, A' and C should be increased by two pipe diameters, and B by eight pipe diameters.

Note 2. Straightening vanes will not reduce lengths of straight pipe A. If straightening vanes are needed because of other fittings, they may be located preceding the reducers (see top sketches); or if they must be located in the straight run of pipe following the swedge, length A is to be increased by an amount equal to the length of the straightening vanes (see bottom sketches).

Note 3. When the diameter of the orifice may require changing to meet different conditions, the lengths of straight pipe should be those required for the maximum orifice to pipe diameter ratio that may be used.

The fourth installation sketch shows the use of a reducer preceding the run of straight pipe. This type of installation makes a rather easy installation insofar as obtaining accuracy is concerned, inasmuch as the minimum lengths of the straight pipe required between the reducer and the orifice flange are quite moderate in amount.

V. Installation with Valve or Regulator Preceding the Orifice Run



Note 1. When "pipe taps" are used, length A, A' and C should be increased by two pipe diameters, and B by eight pipe diameters.

Note 2. Line A, A' and C apply to gate valves which are used for throttling the flow (partially closed).

Note 3. When the diameter of the orifice may require changing to meet different conditions, the lengths of straight pipe should be those required for the maximum orifice to pipe diameter ratio that may be used.

The fifth installation sketch shows a gate valve, or a regulator, located upstream from the orifice. Any valve when used for throttling the flow, produces considerable disturbance that is ironed out only by long runs of straight pipe, or by straightening vanes between the disturbance and the orifice flange. Wide open gate valves may be counted as so much straight pipe.

Lake Water for Precooling

Use of lake water for air precoolers and subsequently in refrigerant condensers is expected to result in marked economies in the operation of the air conditioning system in the Church and Dwight Co. factory, Syracuse, N. Y. The lake water, which reaches a temperature of 40 F or less in winter and seldom more than 65 F in summer, is first supplied to the air precooling coils and then circulated through refrigerant condensers. The control is arranged so that when cooling is required, as much as possible is done with the precooling coils before mechanical refrigeration is brought into operation.

During April, May and June, it is planned to use lake water exclusively, while in the remainder of the warm weather water and mechanical refrigeration will be used.

Air conditioning is particularly needed in this plant to insure cleanliness and purity of product. Humidity control is necessary for the proper operation of the mechanical equipment used in the process. This calls for the filtering of air in certain sections of the plant as well as thorough ventilation and humidity control. The plant is a bicarbonate of soda refining and packing plant.

Automatic humidity controls maintain a correct balance between the mixture of inside and outside air.

Corrosion in Steam Heating Systems

Removal of Gases from Feedwater

By Leo F. Collins* with Everette L. Henderson†

DATA on the removal of oxygen and carbon dioxide from feedwater have been obtained from about seven pieces of plant equipment but only those representative of four installations are recorded. These are believed to typify the performance of equipment in each class. In these four instances a brief description of the unit and its operation are given, so its limitations as well as its accomplishments may be known.

Non-Deaerating Open Feedwater Heater

Open feedwater heaters, while not designed for the explicit purpose of deaeration**, nevertheless do perform this function in a limited way. Since such heaters are by far the most universal type in use it seems the performance of at least one is worth recording.

Description of Heater Tested—The important features of design of the heater tested are shown in Fig. 22. Water is introduced near the top, cascades over baffles while in contact with steam, and then falls into the water space. In the water space the temperature is maintained at about 170 F.

Results Obtained—Many analyses of the effluent from this unit have been made. The data for dissolved oxygen are shown in Fig. 23. The curve represents the theoretical temperature-solubility relationship for water exposed to the normal atmosphere. It is seen from the figure that usually more oxygen is present than for water exposed to the atmosphere at the same temperature; this, however, is to be expected. In the heater tested the water is only mildly agitated and insufficient time is provided for equilibrium to become established. Then, too, the steam used for heating and "blanketing" the water often contains some oxygen.

This unit was never found to expel all the free CO_2 [carbon dioxide] from sodium bicarbonate solutions of even relatively high concentrations.

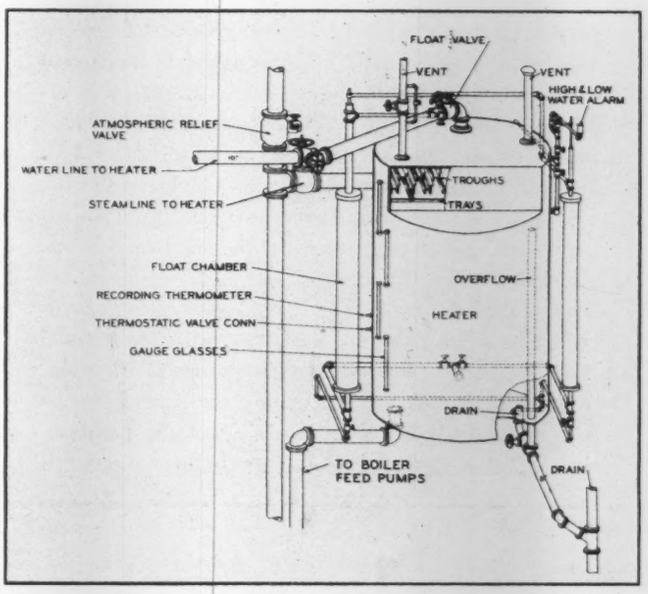


Fig. 22—Non-deaerating feedwater heater

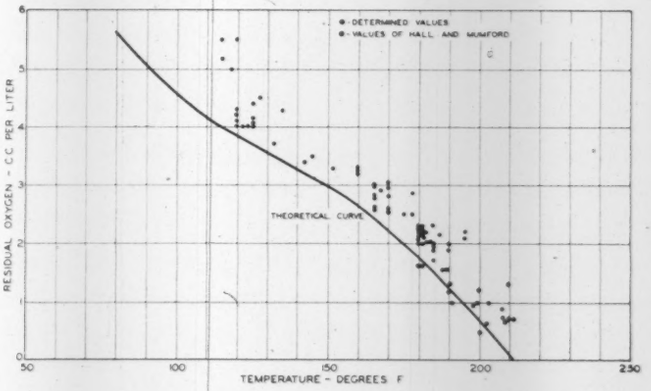


Fig. 23—Oxygen concentration of effluent from a non-deaerating open feedwater heater

Tray Type Deaerating Feedwater Heaters

Data have been obtained for three tray type deaerators but only those pertaining to two will be reviewed. The other unit has since been revised because it failed to approach its expected performance and also because of mechanical troubles. Those considered were designed largely upon the basis of experiences obtained with the first unit.

Description of Units—The two units studied are of identical size and design. A cut-away sketch of one is shown in Fig. 24. Each heater is elliptical in cross-section and is 20 ft long, set with its minor axis horizontal.

There is a vertical diaphragm between each of the tray compartments and a separate vent and vent condenser is provided for each compartment. This makes, in effect, four heaters incorporated in one shell. Baffles are situated so as to direct the flow of steam horizontally through the lower trays and then back through the upper trays.

Water enters through vent condensers and is distributed by suitable piping to troughs (frequently called "water boxes") situated in the top of the vessel. Upon overflowing the troughs it percolates first through a stack of "heating trays" and then cascades over "air

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†Professor of Chemistry, University of Detroit.

**In power plant parlance the removal of extraneous gases from feedwater is usually referred to as "deaeration," although it is quite universally recognized that gases other than air are also removed.

Part 7. Part 1 was published in HEATING, PIPING AND AIR CONDITIONING, September, 1939, pp. 539-542; Part 2 in October, pp. 620-622; Part 3 in November, pp. 675-677; Part 4 in December, pp. 735-738; Part 5 in January, 1940, pp. 24-27; and Part 6 in February, pp. 99-101.

Laboratory experiments bearing upon the removal of carbon dioxide by treatment of the feedwater was the subject of last month's installment of the Collins-Henderson treatise on corrosion in steam heating systems. Data obtained from plant equipment on the removal of oxygen and carbon dioxide from feedwater are presented this month. It has been shown that the concentrations of these two gases must be kept within tolerable limits to prevent excessive corrosion. . . . The comprehensive treatment of heating system corrosion, of which this is a part, is based upon 15 years of research work for The Detroit Edison Co. Engineers concerned with operation and maintenance of plant and building heating systems, district heating engineers, and industrial piping engineers will find the information as useful as will designers and installers of heating plants. . . . Because of its length, it is necessary to publish this material in serial form; the reader is accordingly cautioned not to pass final judgment on the data until publication of the entire article has been completed

separating trays" into the water spaces. The heating trays are horizontal and perforated, the air separating trays are set on edge, ribbed lengthwise, and are solid. These heaters were designed with the expectation they would remove dissolved oxygen in excess of 0.03 cubic centimeters per liter and all "free" carbon dioxide from a zeolite-softened water containing about 40 ppm [parts per million] of bicarbonate, when handling not more than 600,000 lb per hr of water.

Results Obtained—At no time has the expected performance of either unit been obtained at loads in excess of about 350,000 lb per hr. The shaded area in Fig. 25 shows the limits within which the dissolved oxygen concentrations of the effluent usually fall for all load conditions. Typical data for loads below 350,000 lb per hr are shown in Table 10. At higher loads analyses of samples from each tray compartment give results similar to those shown in Table 11. In the columns headed "Oxygen, Cc/L" in Tables 10 and 11 where the values are given as 0, it means that with the Winkler method as ordinarily conducted, no oxygen could be found.

It seems quite certain that imperfect degasification results at high loads because of uneven water distribution on the trays. The latter is conceived as resulting from exceptionally high steam velocities which tend to sweep the water along into localized spots where it descends spasmodically in streams rather than as even distributed droplets or in thin sheets. In some later designs this has been reputedly avoided by directing the flow of steam downward and parallel to the flow of water. The vented steam is removed near the bottom of the vessel. No data have been collected by us from such units.

When optimum water distribution and steam velocities are provided and the flow of steam is properly directed, the heaters studied satisfactorily remove oxygen and free carbon dioxide from solutions containing considerable bicarbonate alkalinity but they do not decompose much of the carbonates of sodium nor do they remove free CO_2 from a bicarbonate solution that has been practically neutralized with acid. A summary of the relationships which were found between the bicar-

Table 10—Analyses Typifying the Performance of the Tray Type Deaerators for Loads Below 350,000 Lb per Hr[a]

	Influent[b]				Effluent					
	Oxygen Cc/L	Bicarbonate (HCO_3) Ppm	Free CO_2 Ppm	pH Value	Oxygen Cc/L	Bicarbonate (HCO_3) Ppm	Carbonate (CO_3) Ppm	Free CO_2 Ppm	pH Value	Date
Saturated at prevailing temperature	39	19	6.3		0	31	4	0	8.3	4- 6-38
	40	20	6.3		0	32	4	0	8.3	4-10-38
	45	24	6.4		0	35	5	0	8.4	4-14-38
	39	24	6.3		0	35	2	0	8.2	4-18-38
	40	20	6.4		0	36	3	0	8.3	4-26-38
	43	17	6.4		0	33	5	0	8.3	5- 3-38
	46	12	6.4		0	33	6	0	8.4	5-10-38
	44	10	6.4		0	34	5	0	8.4	5-18-38
	46	10	6.4		0	33	6	0	8.4	5-26-38
	45	10	6.3		0	43	1	0	8.3	1-12-38
	35	12	6.2		0	35	0	Trace	8.1	10-12-38

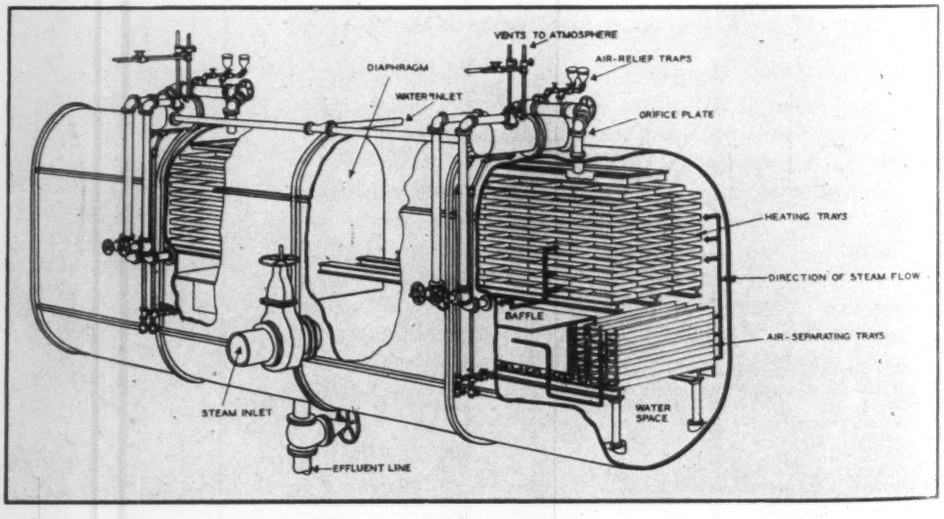
[a]Taken from regular plant control data sheets.
[b]Detroit city water after zeolite softening and partial neutralization with acid.

Table 11—Analyses Typifying the Performance of Different Compartments of the Tray Type Deaerators Under High Load Conditions[a]

Water to Heater M Lb/Hr	1st Comp.			2nd Comp.			3rd Comp.			Back Comp		
	Oxygen Cc/L	Free CO_2 Ppm	pH Value	Oxygen Cc/L	Free CO_2 Ppm	pH Value	Oxygen Cc/L	Free CO_2 Ppm	pH Value	Oxygen Cc/L	Free CO_2 Ppm	pH Value
670	0.36	1	7.5	0	0	8.6	0.16	0	8.7	0.50	1	7.4
684	0.47	1	7.3	0	0	8.5	0.09	0	8.7	0.16	1	7.3
690	0.15	1	7.5	0.25	2	7.1	0.09	0	8.7	0.12	1	7.3
698	0.31	1	7.3	0.28	2	7.2	0.12	0	8.7	0.16	1	7.4
706	0.22	2	7.3	0.34	1	7.3	0.12	0	8.6	0.50	2	7.3
715	0.25	1	7.3	0.28	1	7.3	0.06	0	8.8	0.19	1	7.3

[a]Influent analyses similar to those in Table 10.

Fig. 24—Tray type deaerating feedwater heater



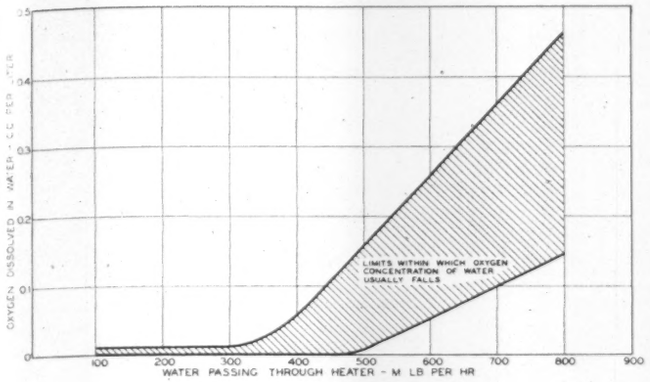


Fig. 25—Limits within which oxygen concentrations fall when water is evenly distributed between tray stacks

Table 12—Relation Between Bicarbonate Content and Free CO_2 of Effluent From a Tray Type Deaerator Under Optimum Operating Conditions

Heater Influent				Heater Effluent			
Bicarbonate (HCO_3) Ppm	Carbonate (CO_3) Ppm	Free CO_2 Ppm	pH Value	Bicarbonate (HCO_3) Ppm	Carbonate (CO_3) Ppm	Free CO_2 Ppm	pH Value
44	0	20	6.3	38	4	0	7.9
13	0	31	5.5	13	0	2	7.3
4	0	40	5.0	4	0	7	5.6

bonate content and free CO_2 for three different degrees of neutrality is shown in Table 12.

Atomizing Type Deaerating Feedwater Heater

Basically the only difference between this heater and those just described is that it agitates the water by the use of spray nozzles rather than trays.

Description of Unit—The important features of design are shown in Fig. 26. In the normal operation of this equipment water is pumped through float controlled valves and vent condensers into four headers situated in the top of the vessel. From the headers it emerges through 16 spring loaded spray nozzles into an atmosphere of steam maintained at a pressure something less than 1.5 lb ga. As a result, the finely divided water is rapidly heated as it gravitates toward the bottom of the vessel. Enroute it is deflected by baffles to a small collecting chamber situated above each of the two atomizing valves. From the collecting chamber the water over-

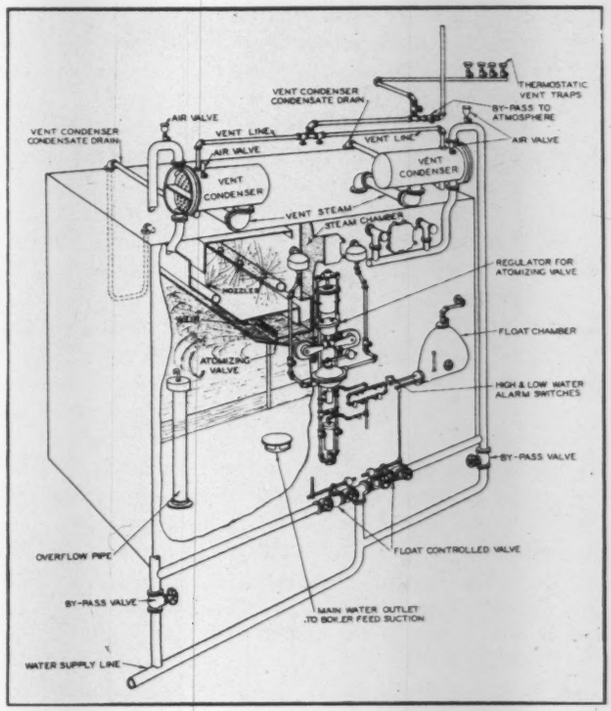


Fig. 26—Atomizing type deaerating feedwater heater

flows into the path of the steam jetting from the atomizing valves. As a result it is violently re-agitated by being blown partly across the heater while in contact with the fresh steam supply.

The flow of steam from the steam chamber is controlled by the atomizing valves. These valves consist of large hinged plates which are operated by a mechanism on the outside of the heater. This mechanism holds the valves partly closed so the pressure in the steam chamber is 0.75 lb higher than in the main space of the heater. Thus, the steam is caused to pass through a restricted opening and a relatively high velocity results. The atomizing valves are the novel feature of this equipment.

The steam, after atomizing the heated water, flows to the top of the vessel and carries along the liberated gases. In the top of the heater it contacts with cold water being discharged from the nozzles. Most of the steam is thus condensed. The remainder condenses in

Table 13—Analyses Typifying the Performance of the Atomizing Type Deaerator Under All Load Conditions

Water Leaving Heater M Lb/Hr	Water to Heater					Water Leaving "Upper Zone"					Water Leaving Heater				
	Oxygen Cc/L	Free CO_2 Ppm	Bicarbonate HCO_3 Ppm	Carbonate CO_3 Ppm	pH Value	Oxygen Cc/L	Free CO_2 Ppm	Bicarbonate HCO_3 Ppm	Carbonate CO_3 Ppm	pH Value	Oxygen Cc/L[a]	Free CO_2 Ppm	Bicarbonate HCO_3 Ppm	Carbonate CO_3 Ppm	pH Value
54	7.5	33	29	0	6.0	0.08	2	29	0	7.4	0	0*	15	6	8.8
87	7.4	36	27	0	5.9	0.14	3	27	0	7.2	0	0	13	6	8.7
108	7.7	32	30	0	6.0	0.11	2	30	0	7.2	0	0	16	6	8.7
117	7.6	34	31	0	6.0	0.14	3	31	0	7.2	0	0	21	4	8.7
162	7.3	34	31	0	6.0	0.20	5	31	0	6.9	0	0	12	8	8.7
189	7.7	20	27	0	6.2	0.20	6	27	0	7.0	0	0	10	4	8.7
204	7.5	30	27	0	6.2	0.20	7	27	0	7.0	0	0	16	6	8.7
243	7.6	31	28	0	6.1	0.31	4	28	0	6.9	0	0	14	6	8.7
273	7.5	32	32	0	6.2	0.81	2	32	0	7.1	0	0	22	4	8.7
											When Only Half of Heater Was Operating				
156	7.5	33	40	0	5.9	No Data Collected					0	0	26	6	8.7
204	7.0	25	35	0	5.7						0	0	21	6	8.7
255	7.5	32	28	0	5.9						0	0	18	4	8.7
305	7.0	34	33	0	5.7						0	0	29	6	8.6
375	7.5	35	33	0	5.7						0	0	29	6	8.7
474	7.5	33	39	0	5.9						0	0	35	2	8.7
510	8.0	20	39	0	6.0						0	0	35	2	8.6

[a]With the Winkler method as ordinarily conducted, no oxygen could be found.

the vent condensers and the non-condensable gases escape to the atmosphere via thermostatic traps.

In the design of this equipment it was expected that a maximum of 803,000 lb per hr of water could be heated, that up to 642,000 lb per hr no free CO_2 would be in the alkaline effluent, and that the oxygen concentration would not exceed 0.03 cc per liter.

Results Obtained—From the foregoing description it will be obvious that some degasification takes place in the "upper zone" and that it reaches a maximum as the water leaves the storage space. In the tests conducted, which embraced a water load (as measured in the effluent line) of from 54,000 lb per hr for the entire heater to a maximum of 510,000 lb per hr with only one-half the unit operating, no effect of load could be determined. Analyses of the effluent were consistent throughout the range noted above.

In Table 13 are recorded some typical analytical data for the influent, the water going to the atomizing valves, and the effluent from the heater.

At no time during these studies could oxygen be found with the ordinary Winkler method of test. Thus it seems safe to report less than 0.01 cc per liter and to conclude that this type of equipment very satisfactorily removes dissolved oxygen; likewise that it will effectively remove free CO_2 from dilute sodium bicarbonate waters. It does not decompose much of the residual bicarbonate, however, and in this respect its performance is comparable to the tray type deaerators previously discussed. Neither does it remove all the free CO_2 from waters otherwise neutral, as is attested by the data in Table 14.

[*To be continued*]

Table 14—CO_2 Removed From a Completely Acidulated Bicarbonate Feedwater by the Atomizing Type Deaerator

	Influent				Effluent				
Sample Number[a]	Bicarbonate (HCO_3) Ppm	Free CO_2 Ppm	Total CO_2 Ppm	pH Value[b]	Bicarbonate (HCO_3) Ppm	Carbonate (CO_3)	Free CO_2 Ppm	Total CO_2 Ppm	pH Value
1	0	56	56	4.0	0	0	9	9	4.0
2	2	56	57	5.4	4	0	9	12	—
3	3	50	52	5.4	—	0	9	—	5.4
4	2	54	55	5.4	2	0	9	10	5.6
5	2	56	57	5.4	2	0	9	10	5.6
6	2	50	51	5.4	3	0	9	11	5.8
7	6	48	52	5.4	4	0	9	11	6.0
8	8	48	54	5.4	6	0	7	12	6.0

[a]Consecutive samples at 10 min intervals. Detroit city zeolite softened water was used.
[b]Determined with calorimetric indicators.
Note: Check analyses showed no oxygen in effluent.

Large Synchronous Motor Drives Air Conditioning Compressor

The largest known synchronous motor powered from a city network for air conditioning has been placed in operation in the new International Building in Rockefeller Center, New York City. Operating at unity power factor, the 1000 hp, 1200 rpm, three phase, 60 cycle, 208 volt motor drives a 4300 rpm centrifugal compressor. The noise level of the unit was 83⅓ decibels during tests.

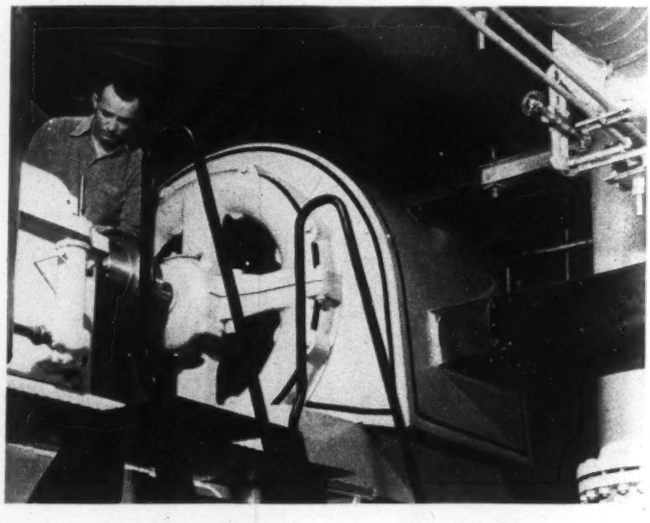
Minimum vibration and noise were secured through special design and correctly proportioned electrical and mechanical parts, its maker states. To illustrate the degree of manufacturing accuracy, the coupling, a fractional part of the rotating element, was pressed on the motor shaft while under construction to balance it accurately and dynamically along the rotor. A steel hood enclosing the back of the motor frame further reduces noise.

Voltage is maintained constant with induction type regulators. The maximum load that may be applied at one time without affecting other connected loads, such as lamps and motors, is 3200 amp. Since the locked rotor or starting current at full voltage is 9500 amp, it is necessary to reduce this value to the set limit. The control keeps the current additions within this value and timed to permit the regulator to restore the normal voltage. The control, essentially a bank of resistors with relays and contactors, is timed and proportioned to limit the current increases and the intervals between additions.

The motor is first connected in series with the full bank of resistors, limiting the total current drawn from the line to 3160 amp. After a predetermined time interval, sufficient to allow the regulator to restore normal voltage, the first section of resistance is shorted out of the circuit, and at this stage 3000 amp is drawn from the line. Again the regulator restores common voltage, and the second step of resistance is short circuited, with 2800 amp drawn from the line at this time. At a later interval, the motor comes completely up to speed, is placed directly across the line and the field is excited. The motor is now synchronized and in regular operation.

During this starting sequence, the motor is gradually brought to synchronous speed, with a consequent falling off in current drawn from the line. Reduction from this cause, in addition to the starting sequence, results in the final decrease to full load current of 2160 amp.

The largest known synchronous motor powered from a city network for air conditioning



Power Consumption and Operating Data for Storage Air Conditioning Presented

By J. M. Dean*

THE refrigeration storage principle in air conditioning is receiving considerable attention these days, particularly from the electric utilities. Power companies know that there are disadvantages associated with storage systems but several of them like the storage idea because it leads to better load factor, offpeak operation and minimum demand during peak periods. It is their conviction that these engineering advantages deserve consideration.

The disadvantages of storage systems are generally considered to be greater space requirement, greater initial cost and greater energy consumption per operating season. These, of course, all concern the owner. Tending to offset these disadvantages are two rather substantial advantages which accrue to the owner. One is low connected horsepower, resulting in lower demand charges on his power bill if he is on a demand rate. The other is dependable performance in extremely hot or humid weather.

The advantages of the storage idea are not always great enough to throw the owning and operating economic balance in favor of a storage system. The most favorable applications for the storage system are those in which the cooling load exists for only a small percentage of the day or week, such as churches, lodge rooms or restaurants serving one meal a day. The advantages of the storage system decrease as the ratio of air conditioning load hours to no-load hours increases. However, power companies are interested in seeing the system made practical for loads of eight to 10 hr duration per day so that it can be more widely applied. Undoubtedly better engineering and mass production can do much to reduce present disadvantages. It probably remains, however, for power companies to contribute, as their share of the effort, some sort of rate compensation for offpeak operation.

The electric utility's overhead costs on power for air conditioning have been rather high because of the large demands and the few hours of operation per year. Furthermore, some electric system peak demands are seriously aggravated by air conditioning loads. Refrigeration loads possess a very small diversity factor during hot days. One can wager that at 10 or 11 a. m. on a real hot day, practically every piece of refrigeration equipment on the system will be operating. This load in Des Moines grew to such a volume that it became an important factor in the need for an expensive extension to the generating plant capacity.

At the end of 1939, Des Moines had 412 air conditioning installations totaling 7697 hp; the average installation is over 18.5 hp. Even omitting two large installations totaling 2446 hp, the average is 12.8 hp per job. This may look like a creditable sales performance in a city of 150,000 population, but to the power company it constitutes an important factor in its operating program.

*Engineer, Des Moines Electric Light Co.

Office Space Air Conditioned

Early in 1939 the Des Moines Electric Light Co. decided to remodel and air condition its engineering and meter departments office space. This project afforded the opportunity to install a storage type system for study. With all the above thoughts in mind, the object of installing this equipment was to attempt to find answers to such questions as:

1) How practical can a storage system be made today?

2) Can such a system produce satisfactory room conditions during a 10 hr cooling day, with a compressor motor not over half the size of that in a conventional system and still have enough non-operating time to satisfy conditions for offpeak periods?

3) What are the actual losses due to adding one or two heat transfers to the overall cycle of the system?

4) What kind of a kwhr per ton per cooling hour factor can be obtained for a partial storage system as compared to a conventional system, particularly for a 10 hr cooling day?

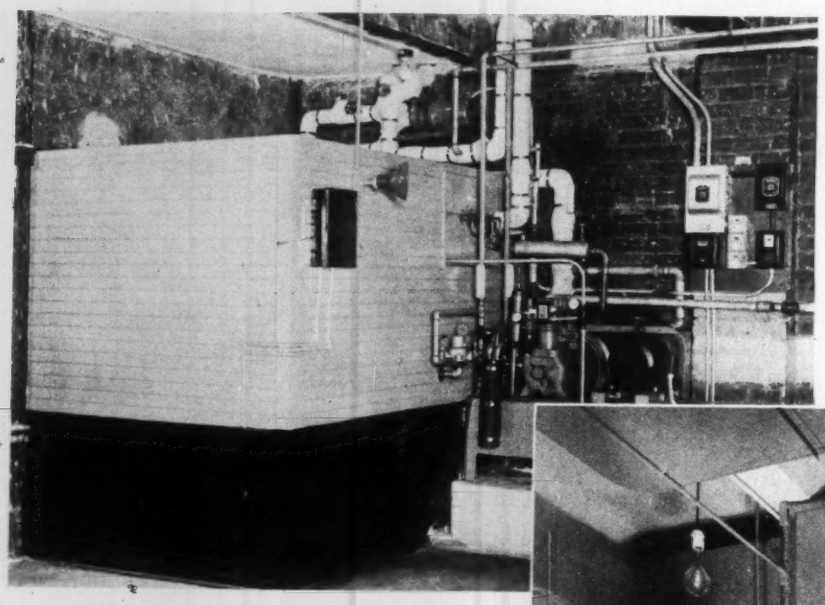
5) Can the losses and extra costs, as found, be offset by offpeak rate compensation?

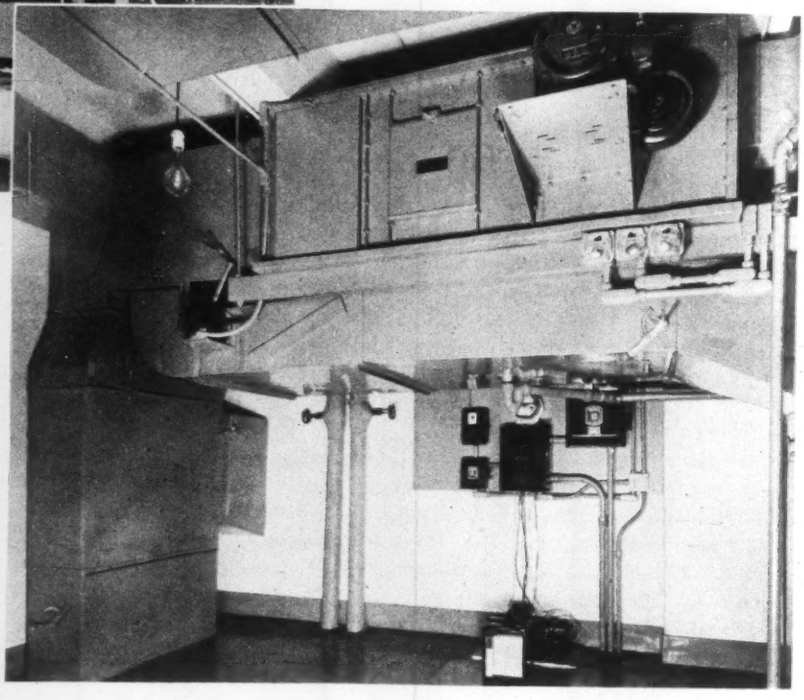
6) If not, can the design of the system be modified to make this compensation possible?

The heat gain for this job was calculated at 10.25 tons on a design of 95 F dry bulb, 78 F wet bulb outside, 80 F dry bulb, 67 F wet bulb inside with a ventilation load of 1.5 air changes per hr. Condensing water was figured at 75 deg. At these conditions the sensible heat component figured 8.26 tons and the latent heat component 1.99 tons.

To insure satisfactory inside conditions at periods of high outside humidity, a secondary design was set up on

When the Des Moines Electric Light Co. decided last year to remodel and air condition its engineering and meter departments office space, an opportunity was afforded to install a storage refrigeration type system for study. A great deal of data were obtained last summer, and the results of this work are reported here. Operation of the storage system is compared with that of conventional systems, and a discussion of offpeak power for air conditioning from the utility standpoint is given . . . . . The author concludes with practical suggestions which should be given consideration when a storage job is designed and installed . . . . . More extensive studies of the same installation — and of a new job — are planned during 1940





The upper view shows the storage tank and condensing unit, and the other picture shows some of the air handling equipment. The gas fired furnace, which appears in the schematic diagram, is not shown in either of these

the basis of 80 F dry bulb, 78 F wet bulb outside and 77 F dry bulb, 64 F wet bulb (49 per cent relative humidity) inside, with a ventilation load of 1.5 air changes per hr. This resulted in a load of 9.07 tons—5.74 tons sensible heat and 3.33 tons latent heat.

The space consists of the second floor of a three story brick building and one room (meter shop) on the third floor. The 13 in. brick walls, 6 in. concrete floor and ceiling were insulated with rock wool, metal lath and plaster, giving a considerable amount of heat lag to the structure.

After a preliminary study of various combinations of equipment, it was decided to design the system for semi-storage, using a "Freon" compressor of 5 tons nominal rating. It was desired to have the compressor operate mostly at night, but it was permissible to let it operate part of the time during the day. The limitation of daytime operation was that it should have capacity enough to permit being shut down 2 to 3 hr during the morning at the period of the system peak.

The diagram and pictures show the general layout of the system. The compressor, storage tank and circulating pump are on the first floor, in a garage room, directly below a small room housing the balance of the equipment. This equipment room on the second floor is adjacent to the conditioned space.

The storage equipment consists of a galvanized iron storage tank (insulated with 6 in. of cork and boarded with car siding) housing a factory made, copper tube ice making evaporator, similar to the one described in the article Storing Refrigeration for Air Conditioning in the June, 1938, HEATING, PIPING AND AIR CONDITIONING. A thermostatic, automatic, three way blending valve is used in the cold water circuit to provide water of constant temperature to the cooling coil. All of the component parts are standard equipment, available from reputable manufacturers.

A rather comprehensive test was run on this system from August 18 to September 16, 1939. Sufficient instruments and appropriate log forms were provided for determining heat absorbed by the evaporator, condenser and circulating water; total electrical and water input to the system; conditions at which compressor operated; compressor, pump and fan hours of operation; and weather conditions and room conditions. A brief summary of the test data is given in Table 1.

Summary of the Test Results

The conclusions to be drawn from this test are necessarily tentative. The data were not analyzed until the completion of the test and an additional test will be run next season after making some adjustments to obtain better performance. However, enough information was secured to give some definite answers to the questions under study:

Practical—The system proved quite practical in several respects. The cooling and comfort results were excellent. The space requirements, though greater than for a conventional system, were not a handicap. The storage tank and compressor were installed in a corner of the garage that was not usable for car storage. It is conceivable that similar waste space could be found for many installations.

Initial Cost—The initial cost was about 25 per cent greater than for a conventional system of the unit type.

However, compared to the cost of a conventional system of the assembled, composite type, the cost difference would not be as much as 25 per cent. It seems reasonable to assume that if all the equipment, except the storage tank and evaporator, could be built into one or two cabinets like unit type conditioners, this cost difference could be reduced to 15 or even 10 per cent.

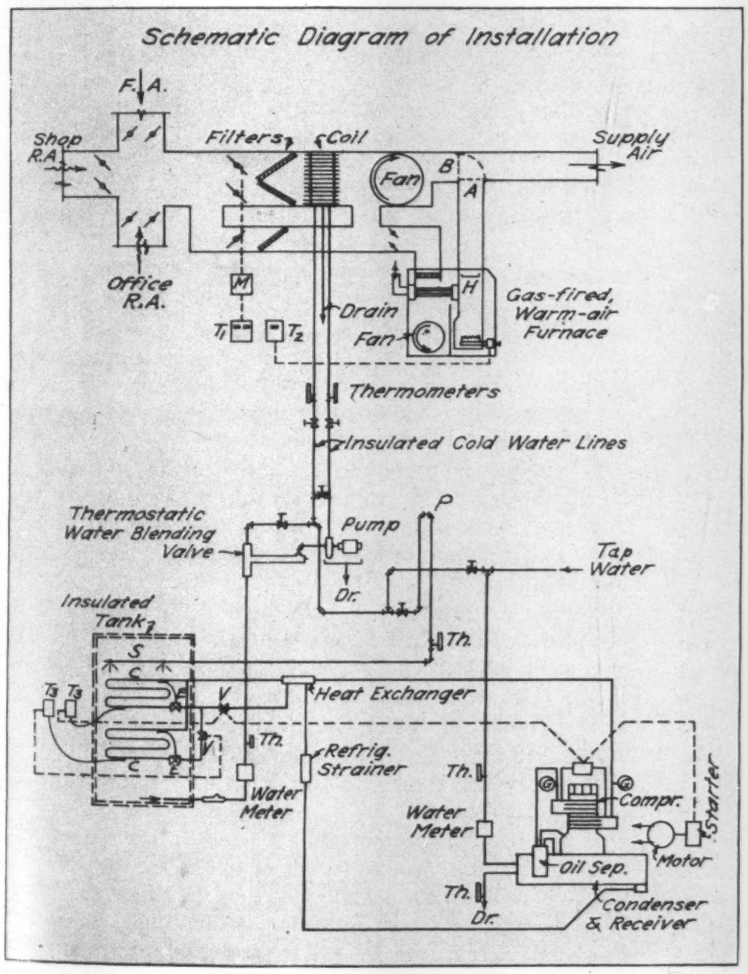
Conditions Maintained—The system not only produced satisfactory room conditions for a 10 hr cooling load (7:30 a. m. to 5:30 p. m.) but, during the test period, the compressor ran only 62.3 per cent of the time, allowing ample leeway for non-operation during periods of system peaks. During seven days of the test the outdoor temperature reached peaks of 95 F or over. On three of these seven days the outdoor temperature rose to 101 and 102 F. The effective temperature in the room during these extreme temperatures was around 73 F.

Demand—The maximum kw demand of such a system is inherently only about 60 per cent of that of a conventional system, for similar application, and yet we here have ample evidence that the demand of the compressor motor could be removed during system peaks, leaving an equipment demand, during peak periods, of about 8 to 10 per cent of that of conventional installations.

Table 1—Summary of Test Data on Storage System

Total hours of test, noon 8/18 to noon 9/16	696
Room cooling load hours (hours of pump operation)	256.57
Compressor hours of operation	434.60
Fan motor hours of operation	289.95
Total kwhr consumed	2,454
Condensing water gallons	354,316[a]
Condensing water Btu extraction	28,041,000
Compressor-evaporator Btu extraction	18,983,300
Circulating cold water Btu extraction	17,509,000
Efficiency (per cent) of tank transfer, 17,509,000/18,983,300	92.2
Kwhr consumed per room cooling hour	9.55
Kwhr per ton per room cooling hour	0.93
Condensing water per room cooling hour, gallons	1,380[a]
Mean effective temperature inside at noon, F	71.78
Mean weighted average values of compressor performance:	
Head pressure, lb per sq in. ga	111
Suction pressure, lb per sq in. ga	18.6
Btu per hr capacity	43,700
Average demands of motors, kw:	
Compressor	5.0
Pump	0.652
Fan	0.775
Total average demand of motors, kw	6.427
Load factor	0.546
Average heat extraction from cooled space, tons per cooling hour	5.7

[a]The water consumption was excessive due to a faulty automatic water valve on the condensing unit. Normal consumption for this test would have been about 149,000 gal, as indicated by the manufacturer's rating table. This amounts to 580 gal per room cooling hour.



Schematic diagram of the installation. A—position of damper for summer use. B—position of damper for winter use. C—ice-making coils. Dr.—drains. E—expansion valves. G—head pressure and suction pressure gages. H—humidifier pan. M—damper motor. P—pump prime-seal loop. S—sprays. T_1—cooling thermostat. T_2—heating thermostat. T_3—coil thermostats. Th.—thermometers. V—solenoid valves. Compressor rating, 5 tons

Heat Transfer Loss—The heat loss due to using water as the intermediate transfer medium was about 7.8 per cent. Such a loss would also exist on a conventional system of the indirect type. This loss was determined by thermometers and a water meter in the cold water line and by the suction and head pressure gages on the compressor. The thermometers, meter and gages were read frequently during the day. The Btu absorption of the water was compared to the compressor-evaporator capacity on the basis of weighted mean averages of the operating pressures. This loss had been estimated by contractors' sales engineers, prior to the installation, at two to three times this.

Power Consumption—The average kwhr per ton per cooling hour for the test was 0.93. Careful analysis of the data indicates that this factor may be reduced to 0.87 under optimum operating conditions. These figures compare favorably with operating cost estimating methods used by several manufacturers of conventional equipment. However, power consumption and cooling hour data were secured from nine local users of conventional systems for periods approximately coinciding with the test period and the average for these nine jobs was 0.61. The kwhr per ton per cooling hour was chosen as the only apparent unit of comparison between the performance of storage and conventional systems.

There is a possibility of error in the above comparison, due to three conditions: (a) The tonnage factor in this unit was, in the case of the storage system, taken as the job tonnage indicated under conditions of maximum design (10.25 tons) whereas, for the conventional systems, the tonnage was taken as the nominal tonnage rating of the compressor equipment because survey data were not available on most of these jobs. (b) It could not be determined whether the surveys for the various installations had been made on the basis of the same maximum design conditions. (c) Test periods were not identical.

There was considerable variation in the kwhr per ton per cooling hour factors for the nine conventional jobs studied:

Application	Period of Data	Compressor Ton Rating	Kwhr per Ton per Cooling Hour
Jewelry store	8-26/9-26-39	5.67[a]	0.605
Restaurant	8-24/9-25-39	15.8[a]	0.420
Restaurant	8- 2/8-31-39	7.5	0.575
General office	8-23/9-23-39	15	1.010
General office	8-23/9-23-39	5	0.555
General office	8-19/9-16-39	15	0.615
General office	8- 2/9-25-39	15	0.700
General office	8- 2/9-25-39	15	0.575
General office	8- 2/9-25-39	10	0.460
Average			0.61

[a]Maximum tonnage per survey.

The average of 0.61, however, seems to be about right as borne out by the following computations:

On this test the kwhr expended per therm (100,000 Btu) capacity of the compressor was 11.45, which is obtained as follows: Compressor hours of operation were 434.6 at an average demand of 5 kw. $434.6 \times 5 = 2173$ kwhr power input to compressor motor during period of test. Compressor heat extraction was 18,983,300 Btu, or 189.83 therms. $2173 \div 189.83 = 11.45$.

At nominal conditions for conventional operation of 113.3 lb head pressure and 35 lb suction pressure, the kwhr per therm may be calculated at 7.64. The kwhr per therm ratio for these two conditions is $7.64/11.45 = 0.67$.

The ratio of kwhr per ton per hr for the average of nine conventional jobs as compared to the storage job under test is $0.61/0.93 = 0.65$.

The corresponding ratio for the storage job under computed optimum conditions is $0.61/0.87 = 0.7$.

Since the kwhr per therm ratio falls between the other two ratios, it is evidence that the kwhr per ton per hr factor of 0.61 is a representative figure for conventional systems.

From this test, it appears that the semi-storage system requires 40 to 50 per cent more energy consumption than the conventional system.

It may be noted here that some manufacturers' and authors' methods of estimating operating costs of conventional systems indicate that kwhr per ton per hr factors in the neighborhood of 0.7 to 0.9 and even higher, might be considered as representative values. There are apparently sizable safety factors in these estimates. If it should be found later that 0.61 is too low a figure for conventional jobs, then the economic ratio of storage to conventional systems (for office application) will improve accordingly.

Offpeak Rate Compensation—However, even on the basis of costs as found, it is possible that many power companies can offer offpeak rate compensation that will pay good returns on the added investment and extra owning costs involved in the storage system. This should be particularly true for power companies that experience daily system peaks of 2 to 3 hr duration in the summer months. With only the fan and pump operating during the peak period (say 9 a. m. to 11:30 a. m.), over 95 per cent of the total consumption may be used during offpeak hours.

The cost of generating, distribution and major substation equipment may be around $250 per kilowatt of capacity. With fixed charges of 12 per cent, the overhead for serving conventional air conditioning equipment on the basis of 1000 hr operation per year, is 3c per kwhr consumed. Except for fuel costs and some minor items, most of this overhead can be eliminated in rates to offpeak air conditioning customers.

This situation has not formerly been encountered in building rate schedules but it appears to be the logical solution to the problem of getting air conditioning load shifted from peak to offpeak time.

When this job was installed, provision was made for the addition of a dry expansion "Freon" coil in the conditioner cabinet. The thought was that after trying the ice storage system, it might be desirable to experiment with a combination set-up, letting the compressor work into the ice making coils at night and into the dry expansion coil (at increased capacity rate) during part of the day. This operation will, no doubt, reduce the kwhr per ton per hr factor, although it will increase the investment in equipment.

The tentative plan for the 1940 season is to repeat this test during May and June, taking advantage of what was learned last summer, and also to take simultaneous operating cost data on the nine or more conventional jobs. After that, it is possible that the dry expansion coil will be added and another test run.

Suggestions for Storage Installations

In an installation of this type, there are several constructional and operating items that should be carefully watched. Some of these items, which were brought out in this job, are:

Install the water circulating pump below the level of the water in the storage tank. Otherwise it may be found difficult to keep the pump primed.

An oil separator is desirable on the refrigerant line to assure maintaining oil in the crank case.

Cork insulation of 6 in. thickness is desirable on the storage tank to minimize losses.

When the motor is first started, it is advisable to check the electrical input. When a compressor and motor are picked for low temperature work, it is easy to overload the motor if operating pressures are not kept just about right.

Certain motors require more careful attention to bearing lubrication than others. On this kind of a job the motor runs long hours and the bearing lubrication should be watched very carefully.

The condenser water consumption should be checked carefully for the first two or three days to guard against excessive water consumption due to a defective valve or restricted tubes.

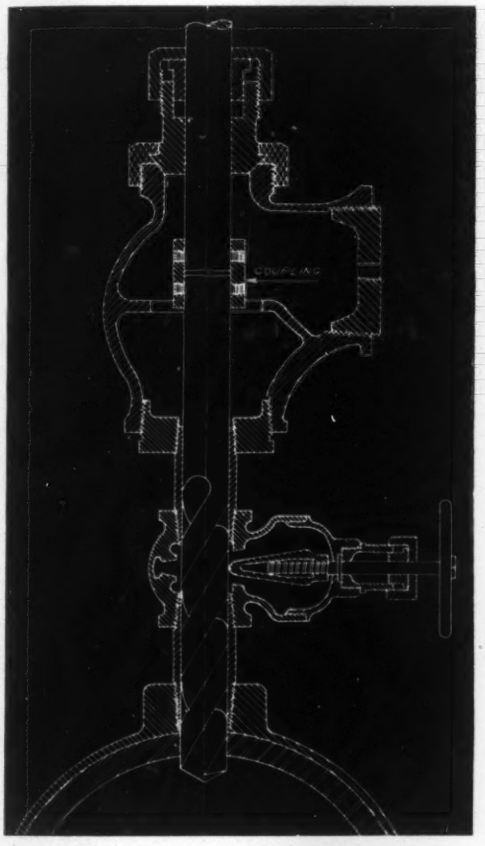
If two control thermostats are used on the ice making evaporators, the capillary tubes of the thermostats should not be parallel in the same conduit or molding; they must be separated to not influence each other, and also insulated.

Practical Piping Problems

Making Branch Connection to Pipe Line Under Pressure

By W. H. Wilson*

PIPE saddles and pipe clamps can often be used to advantage in maintenance work for making a connection to a main pipe line without removing a length of pipe to install a tee in the line, or when it is not advisable to cut into the line with welding equipment. We have used these appliances to install branch connections on



Method of making a branch connection to a pipe line under pressure as used and described by the author

water and fuel oil pipe lines under 50 to 75 lb pressure, the method being as follows:

First apply the pipe clamp or saddle securely to the pipe line at the proper location and mark the exact center of the tapped opening with a center punch (see sketch). Drill into the wall of the pipe with a small drill, taking care not to drill through; this will aid materially in starting the proper size drill.

Install the pipe nipple and gate valve, with the gate valve in the open position. An old angle valve—of a larger size, preferably— can be used with the stem hole drilled out for the shank of the drill so it will pass through the packing box. It may be necessary to weld a rod to the shank of the drill to make it long enough to extend out through the packing box.

The side opening of the angle valve should be provided with a bushing or a plug tapped with a small opening to receive a valve to be used as a bleeder to relieve the pressure after the gate valve is closed. The drill should have a shoulder or stop so that the pressure will not blow the drill through the angle valve. Better results can be obtained by drilling the hole first with a small drill and then with the proper size drill; the small drill will make a lead hole for the larger drill.

After the proper size hole is drilled, draw the drill up into the angle valve body, close the gate valve, open the bleeder valve on the side of the angle valve body, in case there is any leakage through the gate valve that would maintain a pressure, then remove the angle valve body and drilling equipment.

*Member of Board of Consulting and Contributing Editors.

[*Concluded from p. 154*]

valve as regards seats, rings, discs, stem, wheel or general operating characteristics. Advantages claimed for a valve of the construction he described are (1) simplification of manufacturing processes and equipment; (2) saving in weight, which would result in saving in erection labor, material costs, and hangars and supports; (3) savings in pipe line maintenance and cost of pipe line insulation; (4) elimination of possibility of blow holes or porous castings; and (5) possibility of using identical materials in valve bodies and steam lines.

The author showed a specific design for a valve to be made up of two welding reducers, a welding cap cut in half, two special slip-on welding flanges for valve bonnet, two rings to be welded into the large ends of the reducers, to receive the seat rings, two pieces of steel T bar to form the yoke, a piece of flattened pipe to form the portion of the body just below the bonnet, and the necessary small pieces of steel to form glands, collar and guides. It is intended that the steel retaining rings be welded in the ends of the reducers first, the entire valve body then welded together, and the retaining rings tapped and seats placed as in a standard cast valve. Should the service for which the valve is intended require stress relieving or heat treatment, this would be applied before machining. If a wedge seat gate valve is required instead of a parallel seat, the only difference in fabrication would be in the position in which the retaining rings are to be welded.

Albert Upmalis submitted a paper on the Erection of an Air Conditioning Plant in Latvia, in which wide application of the arc welding procedure was made in connection with the construction of the fans, filters, air heaters, and ducts. According to the author, welded construction of the air heaters resulted in a saving of 25 per cent and corresponding savings were shown in connection with the other items.

In addition to the main items of the air conditioning plant, he described the welding procedure as applied also to other parts of it, including ducts, piping, supports, deflectors, etc.

Design of the Piping Service Tunnel at Chronic Disease Hospital Described

By Robert Grahn with T. S. Tenney*

AN interesting example of institutional piping services is to be found at the recently completed hospital for chronic diseases on Welfare Island, New York City. The hospital, which has been open since July 1939, consists of four four-story ward buildings, a five-story administration building, and a three-story laboratory building. These buildings are inter-connected by a corridor running between them on the basement, first, and second floor levels. Two other buildings in the group, the boiler house and the nurses' home, were completed previously.

The boiler house, which is about 700 ft from the nearest ward building, supplies the six buildings of the hospital group with hot water for heating; steam for domestic hot water, sterilizing, and incidental heating; and brine for refrigeration.

The underground connecting tunnel at the new hospital for chronic diseases on Welfare Island, New York City, serves the double purpose of carrying the various piping and other services from the boiler house to the ward buildings, and provides a convenient, clean, lighted and ventilated walkway for the nurses and other personnel. . . . . Details of the tunnel piping are described here. . . . . The buildings are heated with hot water, exhaust steam being used in the converters. There are also some unit heaters supplied with high pressure steam

Service lines are run from the boiler house to the nearest hospital pavilion through an underground tunnel which is about 12 ft wide and 8 ft high. The greater part of the tunnel is a clean, well lighted passageway designed principally for the use of nurses and others walking between the hospital, the boiler house, and the nurses' home. The rest of the space is occupied by two steel pipe racks with a service passageway between them to provide convenient access to the piping for maintenance and repair work.

The pipe racks are constructed of 4 x 3 in. and 5 x 3 in. angle iron pipe rests between 9 in. vertical channel frames. The channel frames are about 10 ft apart. Each of the larger pipes is supported by rollers on the angle iron cross bars. Standard expansion joints are installed in each line.

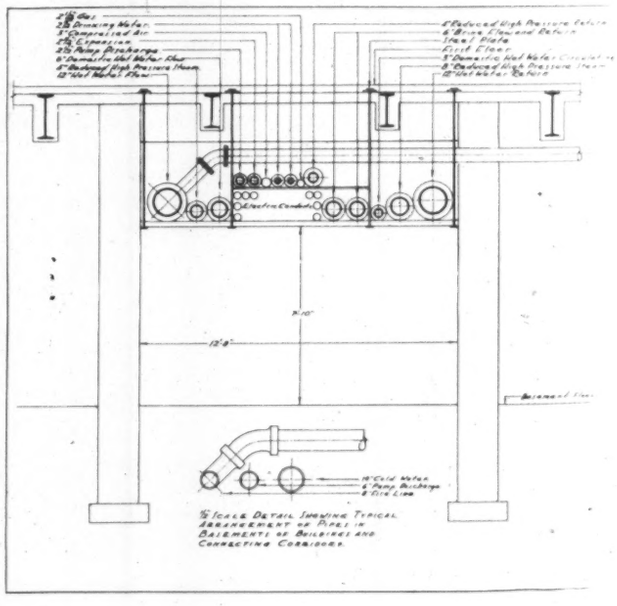
One rack carries a 12 in. hot water heating supply pipe, a 12 in. hot water return, and two 6 in. brine pipes. The other rack carries 8 in. and 5 in. steam lines, a steam return line, pump and trap discharges, a compressed air line, and hot and cold water lines for tunnel cleaning purposes. A 6 in. cold water line, an 8 in. fire line, and a 3 in. drain pipe from the shallow gutter in the service walkway are contained in the floor beneath the pipe racks. Electrical conduits are hung from the ceiling of the walk tunnel.

One feature of the pipe arrangement is the fact that the high temperature pipes are on the inside rack near the upper ventilating windows, as far as possible from the nurses' passageway. This minimizes the direct radiation to the pedestrian portion of the tunnel, which is further protected by the large, intervening heating mains—especially during the summer or non-heating period when they are not in use.

The air intake windows are placed near the floor of the nurses' passageway to aid in keeping this part of the tunnel at a comfortable temperature. The section of the tunnel which is entirely below ground (approximately 550 ft) has 10 of these ventilating shafts and windows.

The designer also considered the possibility of completely isolating the nurses' passageway from the rest of the tunnel with a partition of sheet metal or other thin insulating material secured to the channel supports of the pipe racks. This plan may be easily carried out at any time in the future when it is deemed necessary or desirable.

The distributing lines carried through the tunnel are continued through each of the four ward buildings and the administration building on racks hung from the first floor concrete arches of the buildings and connecting corridors. All supply lines and main return lines rest on double angle iron supports suspended approximately 8 ft above the corridor floors to make for accessibility of



*Tenney & Ohmes, Inc., Consulting Engineers. Member of Board of Consulting and Contributing Editors.

these pipes throughout their entire length. The pipes and their contents weigh about 500 lb per lineal foot. There are expansion loops in the main supply lines at the points where the distributing lines for each building are taken off.

All of the heating is done with hot water, except in a few cases where unit heaters supplied with high pressure steam are used. Two converters are used to bring the heating water to temperatures varying from 100 to 200 F, depending upon the outside temperature. The converters utilize all of the exhaust steam from the steam driven machines in the boiler house. During mild weather this is usually sufficient, but during the winter the converters are supplied for the most part with exhaust steam from the turbines driving the hot water circulating pumps.

This hospital was designed by the associated architects, Butler & Kohn-York & Sawyer in collaboration with the department of hospitals, senior architect, Isadore Rosenfield, and consulting engineers, Tenney & Ohmes, Inc. The heating installation was made by Daniel J. Rice, Inc.

Space Between Floor Joists Used for Heating

Seldom does a city have the opportunity to utilize the same ideas in all of its public buildings. Usually such public structures are built one at a time over a period of years as the need for them becomes apparent. The newest school building in a town may, for example, be equipped with the latest heating and ventilating improvements, while the courthouse, say, built years before, lacks any modern facilities of this kind.

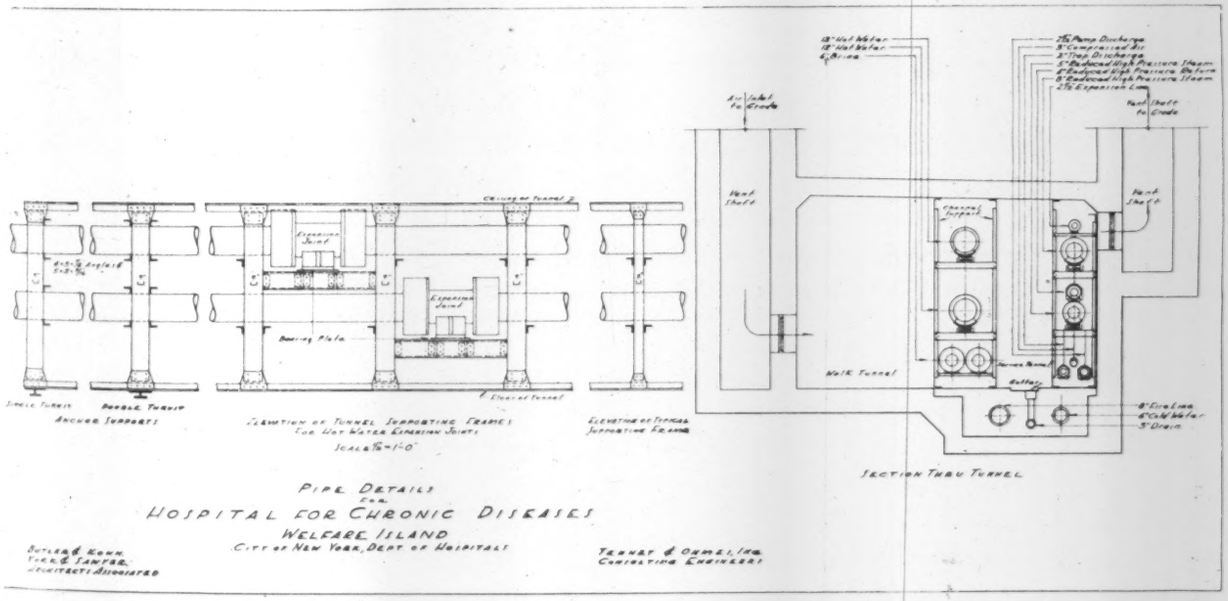
The new Shawneetown, Ill., is a rare exception to the usual community in public building construction. Its courthouse, city hall, grade school and historical museum are being constructed at practically the same time. The old Shawneetown stood for 140 years on the banks of the Ohio River in Gallatin County, but the periodic floods, culminating in the peak inundation of 1937, wrought such heavy destruction that residents voted to relocate the town on a safer site. To accomplish this, the new town was laid out 3½ miles west of the old site, above the crest of the highest flood waters. There the Work Projects Administration and other Federal agencies, together with the State of Illinois and the county of Gallatin are coöperating to create a new community.

Many residences and commercial buildings are being moved from the old site to the new, but new public buildings, as well as a number of homes and business houses are being constructed. WPA labor is being used entirely in the construction of the public buildings.

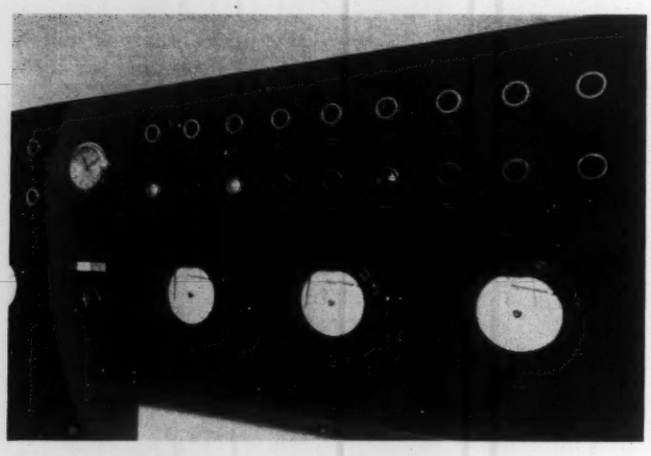
One of the building features is the use of the space between the precast floor joists for heat distribution. The space will be utilized as heat conduits by making a connection with the main heating ducts in the corridors. In designing the heating facilities, the engineers considered the fact that the constant passage in and out of the courthouse, city hall and school would permit frequent entry of cold air into the buildings, necessitating careful planning to keep the structures uniformly warm at a cost that is not excessive. With this in mind the engineers decided to utilize the space between the precast floor joists to distribute heat evenly and economically throughout the entire group of public buildings.

In keeping with the WPA requirement that hand labor be used as much as possible in the construction of the buildings and the use of materials be minimized, the second floor is built of precast joists of I section, with ceiling and floor slabs supported on these joists. This leaves a floor full of concrete ducts between joists. It was decided that these ducts could be used as heating ducts (radiant heat for both first and second stories) by perforating the joist between ducts and by adding insulated supply and return ducts for warm air heating in the corridor ceilings.

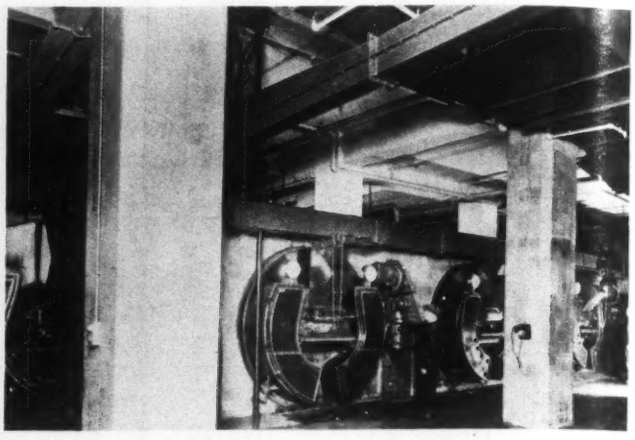
Relocation of Shawneetown above the most extreme high water mark of the flood waters is expected to bring about an improvement in business and industrial conditions. Construction work, except that on private residences and commercial buildings, is being performed by WPA labor taken from the relief rolls of Gallatin County.



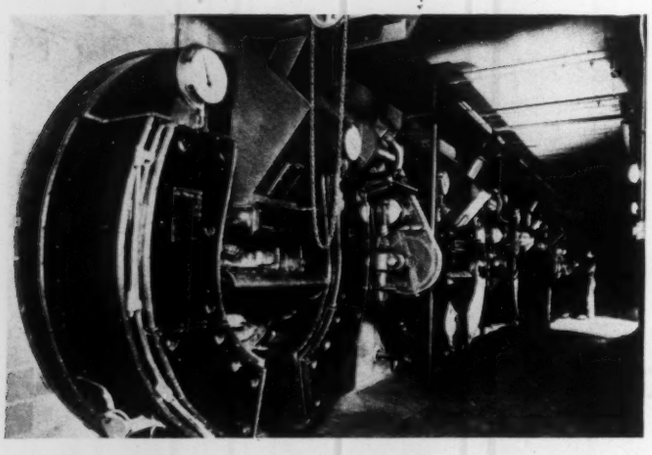
HEATING NEW AIR MINISTRY HEADQUARTERS IN LONDON



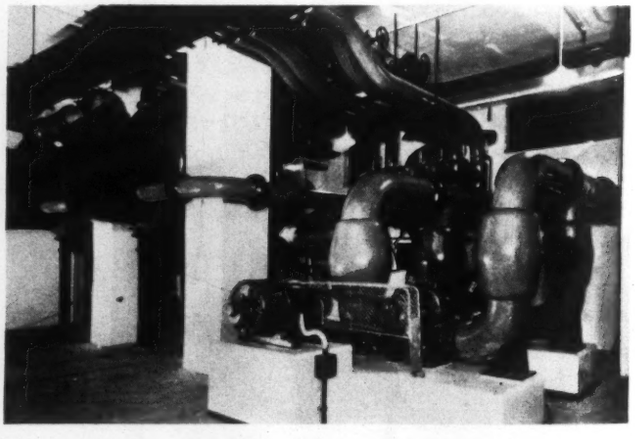
The new Air Ministry Headquarters in Berkeley Square, London, England—which is reputed to be the largest office building in Europe—is heated by a forced circulation, two pipe, upfeed hot water system, with panel type radiators in the offices and floor panels in the main entrance hall. This view shows the control panel of the heating installation. The consulting engineers were J. Stinton Jones and Partners, and the heating was installed by Matthew Hall & Co., Ltd., whose chairman, Mr. B. Baden, furnished the information and photographs given here



The boiler plant supplies heat to the radiator system, the hot water service system, and air heaters on the air conditioning system. Separately controlled circuits are run to the various wings of the building, and the temperatures of these wings may be varied as desired by means of two special mixing valves or blenders. Offices generally are heated by approximately 1400 panel type radiators. Special type concealed floor panels were built into the screed under the marble flooring of the entrance hall, provision being made for expansion and contraction



The four stoker fired boilers have a total output of 20,000,000 Btu per hr. Each stoker can be operated at three speeds, control being by means of pushbuttons mounted on the wall at the side of each boiler. The coal is stored in a large reinforced concrete bunker holding approximately 120 tons, from which it is fed to the boilers by mechanical conveyors. The fuel discharges into the first conveyor through openings in the bottom of the bunker, is elevated to high level and discharged on to a second conveyor, and then to the hoppers on each boiler through chutes



The heating accelerator pumps, which are installed in duplicate, each have a capacity of 66,000 gph of water, which can be reduced to 33,000 gph by means of variable speed motors. All pipework through the building is covered with glass silk; in the boiler room the glass silk is supercoated with cement, and the various pipes are painted to a definite schedule of colors to permit of each service being clearly distinguished. The boilers are also insulated with glass silk, and finished with a planished steel casing, which was specially designed to be easily removable

PANEL TYPE HOT WATER SYSTEM FOR ENGLISH BUILDING

The Psychrometric Chart

Its Application and Theory

By William Goodman*

THERE seems to be some misapprehension as to the method of using the *specific volume* of air in air conditioning computations. When any given air conditioning problem is solved, the weight of air is of primary importance, not the volume. A certain weight of air is required to absorb a given amount of heat. Regardless of variations in its volume, this same weight of air will absorb the same amount of heat for the same temperature rise. On the other hand, the same volume of air will not always absorb the same amount of heat for a given temperature rise—the amount of heat absorbed will depend upon the density of the air. Furthermore, as air is subjected to various conditioning processes, its volume will change, but the weight of dry air in the air-vapor mixture will always be the same. Therefore, the weight of air is not only the most important factor, but the most convenient one for computations because the weight of dry air is a constant quantity.

Specific Volume of Air-Vapor Mixtures

However, when selecting equipment for conditioning air, the weight of air must be converted into volume because equipment is universally rated in terms of volume of air. But, even though the equipment is rated in terms of air volume, the rating is based on standard air having a density of 0.075 lb per cu ft. Hence, in selecting equipment, the weights of air to be conditioned must be converted into volumes of standard air at a density of 0.075 lb per cu ft. Consequently, regardless of the actual temperature and humidity of the air, its actual density should be disregarded and the value of 0.075 lb per cu ft used for determining the volume of air upon which the selection of equipment is to be based. However, if the equipment is actually handling air having a density that differs from 0.075 lb per cu ft, the performance of the equipment must be corrected.

It is necessary to distinguish clearly the fact that the *selection* of equipment is based on *standard* air but corrections for the *performance* of equipment are based on the *actual density* of the air. Thus, the speed of a fan handling air at a density different from that of standard air must be corrected, or the fan will not deliver the required *weight* of air. In the same way, to find the actual brake horsepower required by the fan, the actual density of the air must be known. On the other hand, when selecting a fan to deliver a given weight of air, the selection is based on the density of standard air because the performance curves and ratings of fans are all given in terms of standard air.

The method of computing the actual specific volume of air in any condition was given in an earlier article. (New Tables of the Psychrometric Properties of Air-Vapor Mixtures, HEATING, PIPING AND AIR CONDITIONING, January and February, 1938). Occasionally, it may be necessary to follow the changes in the specific volume of the air on the psychrometric chart. For this purpose, straight lines can be used on Charts A, C and D (p. 359, June, 1939, p. 751, December, and p. 109, February, 1940, respectively) to represent approximately constant values of specific volume. Such straight lines cannot be drawn on the entire field of the high temperature chart, Chart B (p. 750, December, 1939), for reasons that will be apparent later.

Specific volume is defined as the volume of the air-vapor mixture per pound of dry air in the mixture. On a psychrometric chart like Chart A, a line representing the constant specific volume of the air-vapor mixture is very nearly a straight line. As shown in Appendix 7, the slope of a line of constant volume on such a psychrometric chart is

$$q = \frac{0.0329\,Pv}{(0.622 + w)^2} + 854 \qquad [36]$$

A line of constant specific volume on the psychrometric chart must have a slight curvature because the slope is affected by the value of w in the denominator of the above equation. However, for Charts A and C, the range of values of w is small and, hence, the variation in the value of the quantity $(0.622 + w)^2$ is also small. Therefore, an average value of w may be substituted without any marked error. For example, for the middle range chart, Chart A, the value of w varies from zero to 0.022. Substituting the average value of $w = 0.011$, equation 36 reduces to

$$q = 0.0821\,Pv + 854 \qquad [37]$$

The symbol q is used in equations 36 and 37 to represent the slope of the line because the slope of any line

Symbols Used in Text

(For complete list see p. 358, June, 1939, issue)

- P = total barometric pressure, in. of mercury
- p = partial pressure of water vapor, in. of mercury
- q = moisture ratio, Btu per lb of moisture evaporated into the air or condensed from it
- t = dry bulb temperature, deg Fahr
- t' = wet bulb temperature, deg Fahr
- v = specific volume of air-vapor mixture, cu ft per lb of dry air in the mixture
- w = absolute humidity of the air-vapor mixture, lb of moisture per lb of dry air in the mixture

*The Trane Co. Member of Board of Consulting and Contributing Editors.

†Part 10. Part 1 was published in HEATING, PIPING AND AIR CONDITIONING, June, 1939, pp. 357-360; Part 2 in July, pp. 421-424; Part 3 in August, pp. 485-487; Part 4 in September, pp. 549-551; Part 5 in October, pp. 613-615 and p. 617; Part 6 in November, pp. 671-674; Part 7 in December, pp. 749-752; Part 8 in January, 1940, pp. 6-9; and Part 9 in February, pp. 107-110.

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Table 4 - End Points for Wet Bulb Lines
Barometric Pressure 23 in.

Wet Bulb Temp t'	Dry Bulb Temperature, t				Absolute Humidity, w			
	For w=0		For w= 0.022	For w= 0.11	For t=40		For t= 120	For t= 450
	Over Ice	Over Water			Over Ice	Over Water		
1	2	3	4	5	6	7	8	9
-20	-17.70	-17.92						
-19	-16.57	-16.81						
-18	-15.45	-15.69						
-17	-14.31	-14.57						
-16	-13.17	-13.45						
-15	-12.03	-12.31						
-14	-10.87	-11.18						
-13	-9.71	-10.03						
-12	-8.54	-8.88						
-11	-7.37	-7.73						
-10	-6.18	-6.56						
-9	-4.99	-5.39						
-8	-3.78	-4.21						
-7	-2.58	-3.02						
-6	-1.36	-1.83						
-5	-0.12	-0.62						
-4	+1.12	+0.60						
-3	+2.37	+1.82						
-2	+3.63	+3.05						
-1	+4.91	+4.30						
0	6.18	5.54						
1	7.48	6.80						
2	8.79	8.08						
3	10.11	9.37						
4	11.45	10.66						
5	12.80	11.97						
6	14.16	13.29						
7	15.54	14.63						
8	16.94	15.98						
9	18.35	17.35						
10	19.78	18.72						
11	21.24	20.13						
12	22.70	21.53						
13	24.17	22.95						
14	25.69	24.41						
15	27.21	25.87						
16	28.75	27.34						
17	30.32	28.83						
18	31.90	30.35						
19	33.52	31.90						
20	35.15	33.45						
21	36.82	35.03						
22	38.50	36.63						
23	40.21	38.26			0.0000474			
24	41.96	39.91			0.0003826			
25	43.71	41.57			0.0007267	0.0003463		
26	45.52	43.28			0.001082	0.0007249		
27	47.34	44.99			0.001438	0.001105		
28	49.20	46.74			0.001803	0.001494		
29	51.09	48.52			0.002175	0.001890		
30	53.02	50.33			0.002554	0.002293		
31	54.99	52.17			0.002941	0.002705		
32	56.96	54.03			0.003330	0.003120		
33		55.95				0.003550		
34		57.89				0.003985		
35		59.85				0.004427		
36		61.87				0.004880		
37		63.91				0.005341		
38		65.99				0.005810		
39		68.10				0.006289		
40		70.26				0.006778		
41		72.44						
42		74.68						
43		76.95						
44		79.27						
45		81.66						
46		84.08						
47		86.54						
48		89.06						
49		91.63						
50		94.27						
51		96.97						
52		99.68						
53		102.5						
54		105.4						
55		108.2						
56		111.2						
57		114.3						
58		117.4						
59		120.6					0.0001222	

Wet Bulb Temp t'	Dry Bulb Temperature, t				Absolute Humidity, w			
	For w=0		For w= 0.022	For w= 0.11	For t=40		For t= 120	For t= 450
	Over Ice	Over Water			Over Ice	Over Water		
1	2	3	4	5	6	7	8	9
60		123.8					0.0008428	
61		127.1					0.001573	
62		130.6					0.002335	
63		134.1					0.003117	
64		137.6					0.003899	
65		141.3					0.004722	
66		145.0					0.005546	
67		148.8					0.006410	
68		152.8					0.007284	
69		156.7					0.008179	
70		160.9					0.009104	
71		165.0					0.01004	
72		169.3	92.76				0.01102	
73		173.7	97.09				0.01201	
74		178.3	81.54				0.01304	
75		182.9	86.07				0.01409	
76		187.6	90.68				0.01516	
77		192.6	95.54				0.01629	
78		197.6	100.4				0.01741	
79		202.6	105.4				0.01857	
80		207.8	110.5				0.01977	
81		213.3	115.8				0.02101	
82		218.8						
83		224.5						
84		230.2						
85		236.2						
86		242.3						
87		248.6						
88		255.0						
89		261.6						
90		268.4						
91		275.4						
92		282.5						
93		289.8						
94		297.4						
95		305.2						
96		313.0						
97		321.2						
98		329.7						
99		338.2						
100		347.1						
101		356.2						
102		365.7						
103		375.3						
104		385.2						
105		395.3						
106		405.9						
107		416.8						
108		427.8						
109		439.3						
110								0.0002169
111								0.0002654
112								0.0005197
113								0.0007819
114								0.01051
115								0.01330
116								0.01619
117								0.01916
118								0.02213
119								0.02532
120							120.0	0.02866
121							134.1	0.03206
122							148.2	0.03546
123							163.3	0.03913
124							178.4	0.04281
125							194.3	0.04667
126							210.8	0.05071
127							227.6	0.05483
128							244.7	0.05904
129							263.0	0.06351
130							281.9	0.06818
131							301.1	0.07293
132							320.3	0.07767
133							341.2	0.08287
134							362.5	0.08815
135							384.4	0.09362
136							407.0	0.09926
137							430.7	0.10562
138								
139								

Table 5 - End Points for Wet Bulb Lines
Barometric Pressure 24 in.

Wet Bulb Temp t'	Dry Bulb Temperature, t				Absolute Humidity, w			
	For w=0		For w= 0.022	For w= 0.11	For t=40		For t= 120	For t= 450
	Over Ice	Over Water			Over Ice	Over Water		
1	2	3	4	5	6	7	8	9
-20	-17.79	-18.00						
-19	-16.68	-16.90						
-18	-15.55	-15.79						
-17	-14.43	-14.67						
-16	-13.29	-13.55						
-15	-12.15	-12.43						
-14	-11.00	-11.29						
-13	-9.85	-10.16						
-12	-8.68	-9.01						
-11	-7.52	-7.86						
-10	-6.34	-6.70						
-9	-5.16	-5.54						
-8	-3.96	-4.36						
-7	-2.76	-3.19						
-6	-1.55	-2.00						
-5	-0.33	-0.80						
-4	+0.90	+0.40						
-3	+2.14	+1.62						
-2	+3.39	+2.84						
-1	+4.66	+4.07						
0	5.93	5.31						
1	7.21	6.56						
2	8.51	7.82						
3	9.81	9.10						
4	11.14	10.39						
5	12.48	11.68						
6	13.82	12.99						
7	15.19	14.31						
8	16.57	15.65						
9	17.96	17.00						
10	19.37	18.36						
11	20.81	19.74						
12	22.25	21.14						
13	23.71	22.53						
14	25.20	23.97						
15	26.70	25.41						
16	28.22	26.87						
17	29.76	28.34						
18	31.33	29.84						
19	32.91	31.36						
20	34.52	32.89						
21	36.16	34.45						
22	37.81	36.02						
23	39.49	37.62						
24	41.20	39.24			0.0002355			
25	42.93	40.87			0.0005736	0.0001933		
26	44.71	42.55			0.0009219	0.0005649		
27	46.49	44.24			0.001272	0.0009392		
28	48.31	45.96			0.001629	0.001320		
29	50.16	47.70			0.001993	0.001708		
30	52.05	49.48			0.002364	0.002104		
31	53.98	51.29			0.002744	0.002508		
32	55.91	53.11			0.003124	0.002914		
33		54.99				0.003336		
34		56.89				0.003762		
35		58.81				0.004193		
36		60.78				0.004637		
37		62.78				0.005089		
38		64.81				0.005548		
39		66.88				0.006016		
40		68.98				0.006493		
41		71.12						
42		73.31						
43		75.52						
44		77.79						
45		80.12						
46		82.47						
47		84.87						
48		87.33						
49		89.83						
50		92.39						
51		95.02						
52		97.64						
53		100.4						
54		103.1						
55		106.0						
56		108.9						
57		111.9						
58		114.9						
59		118.0						

Wet Bulb Temp t'	Dry Bulb Temperature, t				Absolute Humidity, w			
	For w=0		For w= 0.022	For w= 0.11	For t=40		For t= 120	For t= 450
	Over Ice	Over Water			Over Ice	Over Water		
1	2	3	4	5	6	7	8	9
60		121.1					0.0002374	
61		124.3					0.0009486	
62		127.7					0.001691	
63		131.0					0.002443	
64		134.4					0.003196	
65		138.0					0.003999	
66		141.6					0.004793	
67		145.3					0.005627	
68		149.1					0.006472	
69		153.0					0.007338	
70		156.9					0.008233	
71		161.0					0.009139	
72		165.2					0.01008	
73		169.4					0.01104	
74		173.8	77.20				0.01203	
75		178.2	81.56				0.01304	
76		182.8	86.05				0.01408	
77		187.6	90.70				0.01516	
78		192.3	95.39				0.01625	
79		197.3	100.2				0.01737	
80		202.3	105.1				0.01852	
81		207.5	110.2				0.01971	
82		212.8	115.4				0.02092	
83		218.3						
84		223.8						
85		229.5						
86		235.4						
87		241.4						
88		247.6						
89		254.0						
90		260.5						
91		267.2						
92		274.0						
93		281.1						
94		288.4						
95		295.7						
96		303.3						
97		311.1						
98		319.2						
99		327.5						
100		335.9						
101		344.7						
102		353.6						
103		362.9						
104		372.3						
105		382.0						
106		392.0						
107		402.4						
108		413.0						
109		423.9						
110		435.2						
111		446.6						
112								0.001729
113								0.004219
114								0.006782
115								0.009432
116								0.01217
117								0.01498
118								0.01788
119								0.02086
120								0.02401
121								0.02716
122							127.7	0.03040
123							142.1	0.03389
124							156.5	0.03739
125							171.3	0.04099
126							186.7	0.04476
127							202.9	0.04870
128							219.4	0.05274
129							236.2	0.05686
130							254.0	0.06125
131							272.2	0.06574
132							290.4	0.07022
133							309.9	0.07507
134							330.1	0.08008
135							350.7	0.08519
136							371.9	0.09048
137							394.2	0.09604
138							417.5	0.1019
139							441.8	0.1078

on the psychrometric chart is equal to the *moisture ratio*. Knowing the barometric pressure and the specific volume for which a line of constant specific volume is to be drawn on the psychrometric chart, equation 37 is used to compute the moisture ratio. The slope of the line of constant specific volume can then be found in Table 1 (p. 422, July, 1939, issue). Equation 37 shows that the slope of the line of constant specific volume will vary with both the barometric pressure and the specific volume for which the line is being drawn.

The range of humidities covered by the high temperature chart (Chart B) is too great to permit ignoring the variation in w, the absolute humidity of the air, in the denominator of equation 36. Hence, for a large range in the value of the absolute humidity w, equation 36 cannot be reduced to a straight line equation similar to equation 37 by using an average value of w. Therefore, on this chart, straight lines cannot be used to represent constant volumes, except on the portion of the chart below the absolute humidity line of $w = 0.022$ lb.

Equations 36 and 37 are applicable only to air in the unsaturated and saturated states. The specific volume of fogged air is substantially equal to the specific volume of saturated air at the same temperature. The volume of free liquid suspended in the air affects the specific volume of the air only minutely. For this reason, the specific volume of fogged air may be regarded as constant along any line of constant wet bulb temperature.

Even though the slope of a given line of specific volume is known, it cannot be drawn until some one point is located through which the line can be drawn on the chart. One convenient method of finding the point through which a line of constant specific volume can be drawn is by first locating a line of specific volume for bone dry air; that is, a point along the line $w = 0$. The temperature at which any line of constant specific volume intersects the horizontal line of $w = 0$ can be found from the following equation derived from equation 4 of Appendix 7.

$$t = 1.326\,Pv - 459.7 \quad \ldots\ldots [38]$$

However, some lines of constant specific volume will not intersect the horizontal line of $w = 0$. The following equation, which is also derived from equation 4 of Appendix 7, can be used to find the temperature at which a line of constant specific volume will intersect a horizontal line for any assumed value of the absolute humidity w.

$$t = 0.8244\,\frac{Pv}{(0.622 + w)} - 459.7 \quad \ldots\ldots [39]$$

Humid Air at High Pressure

Charts A, B and C are also useful for computations involving air under high pressures. In order to use the charts for this purpose, several points must be computed on the saturation curve for the pressure in question by means of the following well known equation:

$$w = 0.622\,\frac{p}{P - p} \quad \ldots\ldots [40]$$

Various temperatures are assumed and the vapor pressures of the steam corresponding to these temperatures are found in the steam tables. These vapor pressures are then substituted in the preceding equation. In this equation, P is, of course, the total absolute pressure of the air. After the absolute humidities corresponding to various temperatures have been computed, they can be plotted on the psychrometric chart and the saturation curve for the desired pressure then drawn. After the saturation curve has been drawn, the chart can be used for all computations in exactly the same manner as for computations involving air at ordinary atmospheric pressures.

[*To be continued*]

The method of using the specific volume of air in air conditioning computations is explained this month, as there seems to be some misapprehension on this point. For selecting equipment the weight of air required should be converted into volume of standard air at a density of 0.075 lb per cu ft. Corrections for performance of equipment are based on the actual density of the air to be handled. Use of the psychrometric charts for computations involving air under high pressures is also explained. . . . . Psychrometric charts are fundamental tools of engineers engaged in comfort and process air conditioning work, including drying. The series of which this article is a part includes new charts for the low, medium and high temperature ranges, discussion of the theory underlying them, and instructions for solving types of problems that come up in practice

57 F Ground Water for Condensing

Foundations for the new municipal court building in Philadelphia were installed under a separate contract prior to the award of the contracts for general construction, air conditioning and other branches of the work. During the installation of the foundations, 57 F ground water was found to exist, and steps were taken to make use of it in the air conditioning system.

The refrigerating equipment for the air conditioning system consists of two machines, each of 300 tons capacity. Condensing water will be cooled by two air washer type cooling towers located inside the penthouse. The tests made of the ground water capacity did not indicate that, at the apparently constant temperature of 57 F, there would be sufficient water to supply both refrigerating machines. Pumping equipment was therefore installed of 200 gpm capacity, which, together with a recirculating pump to provide the proper water velocity through the condenser tubes, is sufficient to operate one 300 ton refrigerating machine.

It is anticipated that the average refrigerating load will be such that the ground water will be sufficient for condensing purposes during a large part of the time. No reduction in the capacity of the cooling tower equipment was made because of the use of the ground water. The full refrigerating capacity can be developed, using the cooling towers, in the event of failure of the ground water supply due to seasonal variations in flow, or diversion of the flow by other building construction in the vicinity.

W. R. Morton Keast was the architect for the building, and the consulting engineers were Moody & Hutchison.

Eleven Ways Heating Has Been Improved

By Lewis W. Mauger*

1 In tall buildings there is a considerable chimney effect which tends to draw air into the lower sections of the structure and to lose heated air in the upper portion. In a New York residential hotel this condition became so serious that the lower floors could not be properly warmed and the upper floors could obtain no fresh air except when high winds were blowing. It was necessary to weatherstrip the doors leading from the elevator corridors to the apartments on each floor to prevent this flow of air in the building and to obtain a degree of comfort.

2 Two years ago an apartment house was erected with large metal casement windows in the modern manner; the first cold winter day brought a deluge of complaints from tenants that the apartments were freezing. This extreme exposure may not be duplicated in many buildings but it illustrates the heat loss that can be sustained through windows that are not tight. Metal weatherstrips were the answer to this problem.

3 Heating systems without vacuum pumps must have some means of eliminating air, and the more rapidly it is eliminated the quicker the steam circulation becomes. Small apartment buildings in which it took nearly an hour to force 5 lb of steam into all of the radiators were equipped last spring with a special type of quick vent valve at the tops of risers, after which complete circulation was obtained in about 15 min with less than ½ lb steam. Buildings equipped with vacuum traps should have a regular schedule of trap inspection. Defective trap elements may either remain open, allowing steam to blow through into the return lines, or they may remain closed, permitting no condensate to pass. If steam blows into the return lines, it can reach the traps of other radiators on the same line and cause them to close prematurely, preventing these radiators from heating.

4 Steam under high vacuum occupies a great volume and its condensing temperature is low. It is possible to keep the heating system filled completely with steam at any temperature desired by varying the vacuum setting. This type of system has just been installed in an apartment on Sutton Place, New York City, the high vacuum being obtained by a steam turbine driven pump. In the mild weather we had last fall the exhaust from the turbine has been sufficient to keep the system warm with about 25 in. of vacuum showing on the gage.

5 Often a reduction in expenses can be made by combining the plants of two adjoining buildings, assuming that one is of sufficient capacity to supply steam to the other. In two office buildings steam lines were put through a 42 in. brick foundation wall and an access door was cut through a 36 in. wall. The saving in fuel and labor the first year amounted to about $10,000.

6 Buildings erected 20 or 30 years ago were frequently equipped with high pressure boilers, sometimes used to operate pumps and generators in addition to heating the buildings. In most cities high pressure boilers require continuous attention by licensed engineers and it is often more economical to electrify steam driven equipment and reduce the boiler pressure to that required for heating. This has been done in a number of plants, one of the large printing buildings being a recent example. An investment in four electric pumps and piping changes which ran into thousands of dollars paid for itself in about two years.

7 In setting oil burners or stokers, it is important that sufficient space be allowed for complete combustion of the gases before they enter the tubes. Many boilers require pits to accommodate either stokers or oil burners. One of our buildings was equipped with stokers without adequate pits and the first row of fire tubes was only a few inches above the normal fire bed level. Complete combustion was impossible in the space allowed and tests showed that the CO_2 content of the flue gas ran between 5 and 7 per cent, whereas it should have been double this amount.

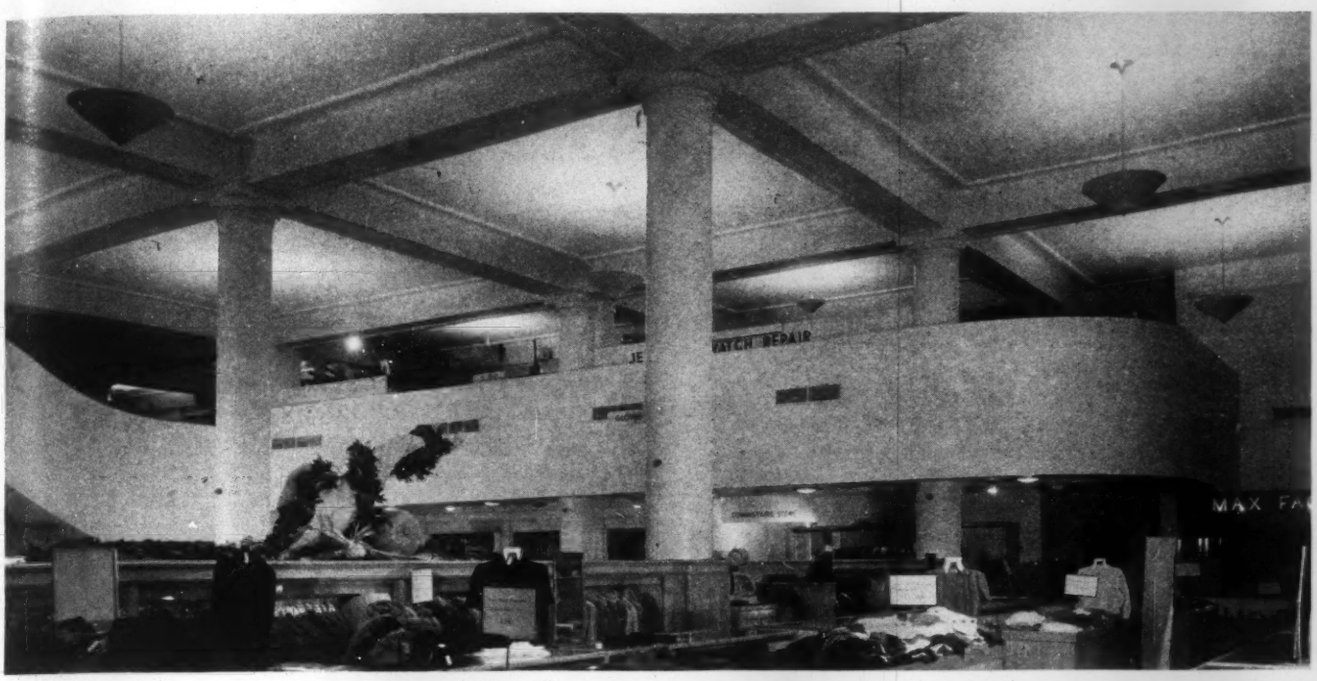
8 Oil burners installed in too small combustion chambers will burn out the refractories rapidly, operate at low efficiency, deposit carbon on the brick work and sometimes soot up the boiler and cause smoke. I am thinking of one case in which the brick work has been renewed each year. Since it would be expensive to rebuild the pit at the present time, we are experimenting with a special brick made to resist high temperature oil flames with the hope that at least two years' service can be obtained. A well designed combustion chamber should last at least three years and then require only minor repairs.

9 In a tall apartment a barometric damper saved about 10 per cent in coal consumption and in another about 15 per cent was saved. A barometric damper should not be installed unless the draft is greater than required for combustion, otherwise the operation of the plant will be stifled. A reliable pressure regulator should be standard equipment in every boiler, especially so when firemen are not posted continuously in the boiler room. I have walked into plants where the firemen have been detailed to make repairs upstairs or in which they were acting as service elevator operators on the side and have found 8 or 10 lb pressure on the boiler when 2 or 3 lb would be sufficient.

10 Where it is desired to burn a low grade of coal without smoking and with a minimum of attention, an automatic stoker often answers the problem. A stoker will often operate at more than 100 per cent boiler rating and is therefore useful in cases where the heating load is greater than the boiler was designed to carry. A Fifth Ave. office building is a good example of this. Two stories were added to the building after it was completed and the only fuel that was capable of steaming the boiler was large size coke. A stoker was set under the boiler a few years ago to burn soft coal and there has been no difficulty in keeping the building warm and only one smoke complaint during the entire period.

11 In larger plants it pays to have meters to check the boiler operation. Meters may be obtained to register or record the draft, CO_2, flue gas temperature, the outside temperature, the inside temperature, the flow of fuel oil and similar items. In plants where permanent meters would not be justified, tests should be made from time to time to determine at least the first three of these items, namely, draft, CO_2 and flue gas temperature. These three items are definitely inter-related and maximum efficiency can be obtained by adjusting them properly. A few years ago, we made a number of tests in various plants and found a surprising number operating with a CO_2 content of between 5 and 7 per cent, which meant that these plants were wasting nearly one-fifth of the coal or oil burned. Changes in firing methods, such as increasing the depth of the fire bed or maintaining level fires without dead spots, adjusting drafts and, in some cases, changing the type of coal brought most of these plants up to desired efficiencies.

*Supervising Engineer, Brown, Wheelock, Harris, Stevens, Inc., Real Estate.



Air conditioning improved the appearance of the main floor. An antiquated balcony rail was replaced with a modern one housing the air ducts, as shown here

Modernize Department Store With Zoned Heating, Air Conditioning

By A. E. Elgar*

WITH the advent of air conditioning in our store last summer, our employees have been able to work in more comfortable surroundings and therefore have been able to do a more efficient job for us. Our ready-to-wear floor definitely had an increase in business. This may well be attributed to air conditioning, which makes the trying on of dresses an easier task in the summer than with the old, stuffy fitting rooms. It can also be said that the store in general benefited from air conditioning, for we noticed an increase in traffic. However, it is hard to determine just how much of this increase was due to air conditioning itself as we remodeled all the floors as well. It is really a combination of the two improvements which resulted in better business.

We are fortunate in having a modern installation which hits a happy medium between the outdoor temperature and the inside cooled temperature, so that when one leaves the store there is not too much of a let-down. We also noted that there were less complaints of "cold air" than we had anticipated. This is due, we believe, to the careful control of the inside temperature in relation to that outside.

Air conditioning has improved the appearance of our main floor balcony. We eliminated an antiquated rail and installed a modern one in its place which houses the air ducts. We not only gained a finer appearance of the floor, but were able to procure a few valuable feet of floor space on our balcony.

Our air conditioning system has also greatly facilitated our heating problem, and made the heating easier to control.

Careful Planning Necessary

As the building is about 35 years old, careful planning was necessary in order to utilize some of the existing equipment, such as duct work and old air washers. The air conditioning plant was designed to serve five floors and the basement.

The refrigerating unit is a 480 ton centrifugal machine driven by a 500 hp condensing steam turbine. Condensing water from the induced draft cooling tower first enters the refrigerant condenser and then goes to the steam condenser. The condensing water is chemically treated to maintain a pH of 8.4, and a daily check is made with a pH comparator.

Chilled water at 45 F leaving the cooler is circulated by two 600 gpm pumps to the 10 cooling and dehumidifying units, situated in the store as follows:

Two on the fifth floor, one serving the fifth floor and the other the fourth.

*General Manager, James McCreery & Co.

The 35 year old building of the James McCreery & Co. department store in New York City was modernized last year with air conditioning and zoned heating. The improvements are given credit for increased business, better appearance and other advantages. . . . Results of the modernization, and the air conditioning and heating, are briefly described here

Two on the third floor, one serving the third floor and one the second.

Three units for the main floor and basement.

One small unit in beauty salon of annex building.

One unit in beauty salon of main building third floor.

One unit in "budget" beauty salon in basement.

Relative humidity in the store is maintained at 55 per cent, and the temperature is varied with the outside temperature according to the following schedule: 95 F outside, 85 F inside; 90 F outside, 82½ F inside; 85 F outside, 80 F inside; 80 F outside, 77½ F inside; 75 F outside, 75 F inside.

The turbine has the following safety controls: (1) Condenser high pressure cut-out. (2) Non-freeze control to cooler. (3) Low oil pressure cut-out. (4) Low refrigerant cut-out. (5) Low chill water flow cut-out (differential flow control). All control switches are closed normally when conditions are not dangerous, and open when dangerous operating conditions exist. They are all connected in series. Failure of power or operation of any one safety control will break the holding circuit and slow down the turbine to a minimum speed.

Heating System Zoned

During the winter months the valves connecting the cooler are shut off, and those connecting the chilled water system are opened to a heater. Hot water is then circulated through the coils. Humidifying is controlled by a hygrostat in the return duct. This governs steam coils submerged in a water pan on the fan side of the coils. The heater for hot water circulation in coils was not designed to give sufficient heat to meet all the requirements of the five floors and basement, but operates in conjunction with our heating system.

Last year zone control was incorporated in our present heating system. At that time thermostats were situated on the fifth floor. Because of the addition of the hot water coils in the fan rooms, these were moved to the eighth floor.

The control board consists of three indicating thermometers showing the temperatures in the three zones. A five point control for each zone is marked "OFF—¼—½—¾—FULL," and also for full automatic. The thermostats on the eighth floor are connected with a time switch. This is set for non-business hours and will maintain a minimum temperature during those hours. At the hour designated to prepare the building for a comfortable temperature it automatically turns on steam full until the building is up to required temperature. Steam flow is then automatically controlled by outside temperature or may be controlled by manually operating the five point controls.

All radiator traps were also removed from radiators and orifices were inserted in the radiator valves, the system having a gravity return.

This daily log form is an aid to efficient operation

MC 442

Daily Log - Air Conditioning

JAMES McCREERY & CO.

Date _____ Machine Started _____ Shut Down _____ Mean Outdoor Temp. _____

" " Wet " _____

Meter Reading:

Steam Start _____ Shut Down _____ Consumption _____ Cost _____

Elec. A. C. Start - _____ Shut Down _____ Consumption _____ _____

Elec. A. C. Start - _____ Shut Down _____ Consumption _____ Cost _____

Elec. D. C. Start - _____ Shut Down _____ Consumption _____

Tower Water _____ Shut Down _____ Consumption _____ Cost _____

Comp. Water _____ Shut Down _____ Consumption _____ Cost _____

Time	Cooler				Condenser				Oil Suction	Oil Discharge	Oil Well	Compressor				Oil Pressure			Cond. Water Tower	No. of Chill Water Pumps	No. of Tower Pumps	Turbine Vacuum
	Vacuum	Liquor Temp.	Water Temp. In	Water Temp. Out	Pressure or Vacuum	Gas Temp.	Water Temp. In	Water Temp. Out				Speed	Steam Pressure	Bearings Suc. Temp.	Bearings Disc	Turbine Oil	Oil Res.	Seal				

What the Operating Engineer Should Know

Water Treatment for Heating Boilers

By Kalman Steiner* and Ralph A. Nelson*

THE general subject of water purification is an important division of chemical engineering; that portion of it concerned with boiler water treatment is of particular interest to men who operate boilers. Large boiler plants require considerable attention to the matter of feedwater treatment in order to assure safe, economical, and continuous boiler operation. In smaller plants the subject is often disregarded entirely, or at least until such time as some operating difficulty focuses attention on it. Probably a better understanding of the subject by those having small plants in their charge would result in minimized plant maintenance, and probably also a not inconsiderable fuel saving. Certainly, it will repay what little effort is required in a small plant to safeguard against scaling and corrosion in the boiler, by taking the steps to assure clean boiler tubes and to avert shutdowns resulting from damaged tubes.

Any consideration of this subject of course involves some chemistry. This need not dismay those unfamiliar with chemistry, for the number of chemical substances involved is small, and the reactions between them can be expressed in simple terms, even though the actual reactions may be complex. We are concerned only with results—not with purely theoretical aspects.

How Water Is Contaminated

Practically all water supplies contain dissolved mineral matter. Only distilled water can be considered as containing no dissolved solids, and even it carries in solution such gases as oxygen and carbon dioxide, and often is contaminated with oil and grease picked up in engines and condensers. Rain water takes up various impurities in its descent through the atmosphere—such as oxygen, carbon dioxide, dust, acids, and organic matter. Surface water from lakes and streams, and underground water from wells, dissolves mineral matter during passage over or through the ground. Water has been called the "universal solvent," because it will dissolve to a greater or less extent a wide variety of different substances.

In addition to dissolved substances, many water supplies also carry some solids in suspension. It is important to have in mind clearly the distinction between *dissolved* matter and *suspended* matter. Most substances dissolved in natural water supplies leave no visible evidence of the fact, as is illustrated by stirring a bit of table salt into a glass of water. There is no change in the physical appearance of the water, but the salt disappears, and no amount of filtration will remove a single particle of it from solution. But if a bit of marble dust be stirred into water, the water becomes turbid—a clearly visible physical phenomenon—and the marble may be removed from the water by passing the water through a filter. To remove dissolved substances from solution requires chemical changes, but suspended matter usually requires only filtration.

The science of water treatment includes the chemical changes employed to remove undesirable dissolved minerals from solution and suspended matter from suspension. Before an investigation can be made into either the facts which make certain minerals undesirable in boiler feedwater, or the procedure for accomplishing their removal, a few simple chemical principles must be understood.

Types of Chemical Substances

Throughout this discussion four *types* of chemical compounds or substances will be discussed: *oxides, acids, hydroxides* (alkalies) and *salts*. An *oxide* is a compound or combination of the element oxygen combined with another element: common oxides are quicklime or calcium oxide, CaO; iron rust, Fe_2O_3; carbon dioxide, CO_2, etc. An *acid* may be defined as a compound of hydrogen with either a non-metal or an oxide of a non-metal, as hydrogen and chlorine in hydrochloric acid, HCl; hydrogen and oxide of sulphur in sulphuric acid, H_2SO_4; and hydrogen and oxide of carbon in carbonic acid, H_2CO_3. An *hydroxide* is best described as the addition product of water, H_2O, combining with a metallic oxide; thus quicklime when mixed with water gives slaked lime or calcium hydroxide, $Ca(OH)_2$, and ferric oxide (iron rust) combines with water to form ferric hydroxide, $Fe(OH)_3$. Finally, acids and hydroxides (alkalies) when mixed together neutralize one another in a reaction whereby the H of the acid combines with the OH of the hydroxide to form water, H_2O, while the metal of the hydroxide combines with the remainder

This explanation of why boiler water treatment is necessary is addressed particularly to engineers concerned with the operation of smaller boilers and heating plants. The authors review—in a simple manner—the chemistry involved, and indicate briefly the effect of various impurities on boiler operation and maintenance. . . . A second article to appear in a later issue of HPAC will go further into the subject

*Ace Engineering Co.

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Elec. D. C. Start - _____ Shut Down _____ Consumption _____

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Comp. Water _____ Shut Down _____ Consumption _____ Cost _____

Time	Cooler				Condenser				Oil Suction	Oil Discharge	Oil Well	Compressor				Oil Pressure			Cond. Water Tower	No. of Chill Water Pumps	No. of Tower Pumps	Turbine Vacuum
	Vacuum	Liquor Temp.	Water Temp. In	Water Temp. Out	Pressure or Vacuum	Gas Temp.	Water Temp. In	Water Temp. Out				Speed	Steam Pressure	Bearings Suc. Temp.	Bearings Disc	Turbine Oil	Oil Res.	Seal				

What the Operating Engineer Should Know

Water Treatment for Heating Boilers

By Kalman Steiner* and Ralph A. Nelson*

THE general subject of water purification is an important division of chemical engineering; that portion of it concerned with boiler water treatment is of particular interest to men who operate boilers. Large boiler plants require considerable attention to the matter of feedwater treatment in order to assure safe, economical, and continuous boiler operation. In smaller plants the subject is often disregarded entirely, or at least until such time as some operating difficulty focuses attention on it. Probably a better understanding of the subject by those having small plants in their charge would result in minimized plant maintenance, and probably also a not inconsiderable fuel saving. Certainly, it will repay what little effort is required in a small plant to safeguard against scaling and corrosion in the boiler, by taking the steps to assure clean boiler tubes and to avert shutdowns resulting from damaged tubes.

Any consideration of this subject of course involves some chemistry. This need not dismay those unfamiliar with chemistry, for the number of chemical substances involved is small, and the reactions between them can be expressed in simple terms, even though the actual reactions may be complex. We are concerned only with results—not with purely theoretical aspects.

How Water Is Contaminated

Practically all water supplies contain dissolved mineral matter. Only distilled water can be considered as containing no dissolved solids, and even it carries in solution such gases as oxygen and carbon dioxide, and often is contaminated with oil and grease picked up in engines and condensers. Rain water takes up various impurities in its descent through the atmosphere—such as oxygen, carbon dioxide, dust, acids, and organic matter. Surface water from lakes and streams, and underground water from wells, dissolves mineral matter during passage over or through the ground. Water has been called the "universal solvent," because it will dissolve to a greater or less extent a wide variety of different substances.

In addition to dissolved substances, many water supplies also carry some solids in suspension. It is important to have in mind clearly the distinction between *dissolved* matter and *suspended* matter. Most substances dissolved in natural water supplies leave no visible evidence of the fact, as is illustrated by stirring a bit of table salt into a glass of water. There is no change in the physical appearance of the water, but the salt disappears, and no amount of filtration will remove a single particle of it from solution. But if a bit of marble dust be stirred into water, the water becomes turbid—a clearly visible physical phenomenon—and the marble may be removed from the water by passing the water through a filter. To remove dissolved substances from solution requires chemical changes, but suspended matter usually requires only filtration.

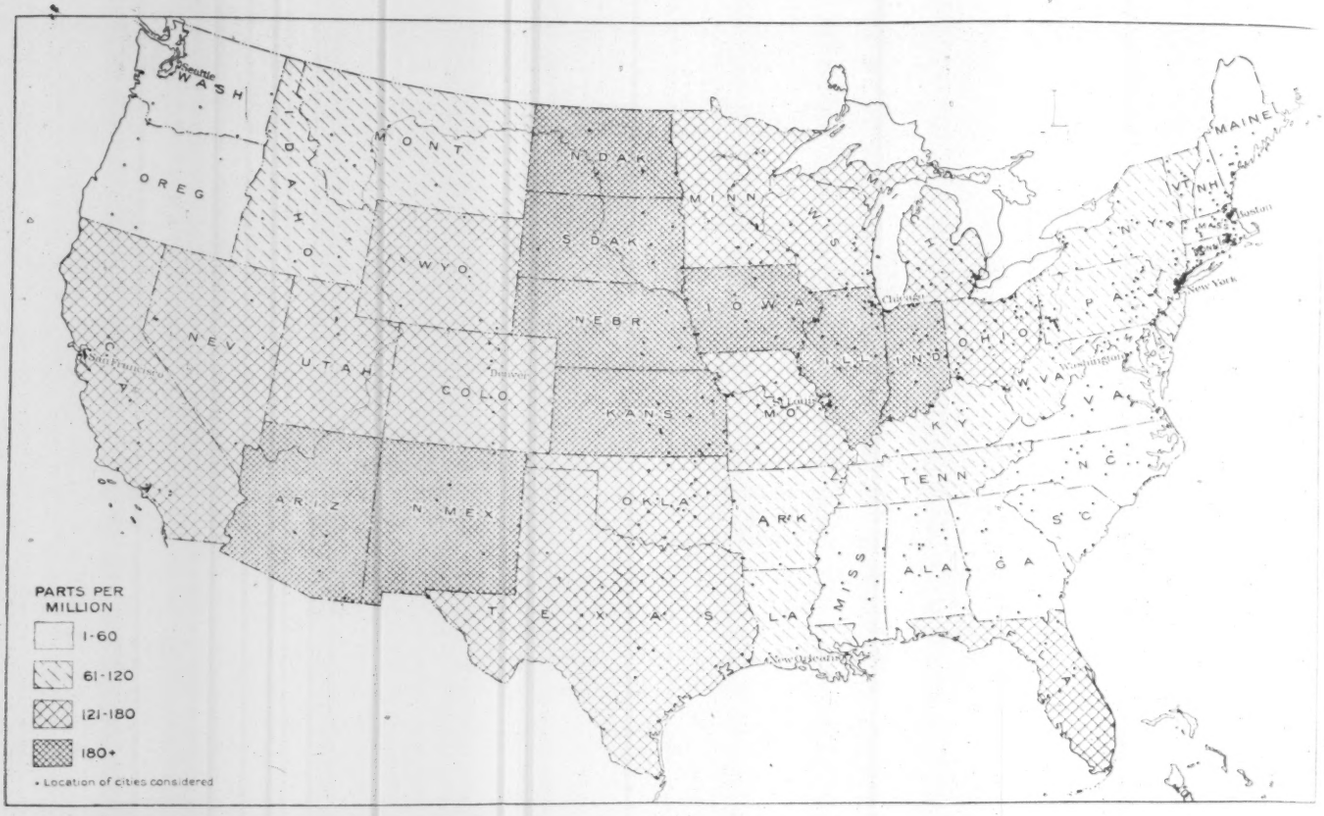
The science of water treatment includes the chemical changes employed to remove undesirable dissolved minerals from solution and suspended matter from suspension. Before an investigation can be made into either the facts which make certain minerals undesirable in boiler feedwater, or the procedure for accomplishing their removal, a few simple chemical principles must be understood.

Types of Chemical Substances

Throughout this discussion four *types* of chemical compounds or substances will be discussed: *oxides, acids, hydroxides* (alkalies) and *salts*. An *oxide* is a compound or combination of the element oxygen combined with another element: common oxides are quicklime or calcium oxide, CaO; iron rust, Fe_2O_3; carbon dioxide, CO_2, etc. An *acid* may be defined as a compound of hydrogen with either a non-metal or an oxide of a non-metal, as hydrogen and chlorine in hydrochloric acid, HCl; hydrogen and oxide of sulphur in sulphuric acid, H_2SO_4; and hydrogen and oxide of carbon in carbonic acid, H_2CO_3. An *hydroxide* is best described as the addition product of water, H_2O, combining with a metallic oxide; thus quicklime when mixed with water gives slaked lime or calcium hydroxide, $Ca(OH)_2$, and ferric oxide (iron rust) combines with water to form ferric hydroxide, $Fe(OH)_3$. Finally, acids and hydroxides (alkalies) when mixed together neutralize one another in a reaction whereby the H of the acid combines with the OH of the hydroxide to form water, H_2O, while the metal of the hydroxide combines with the remainder

This explanation of why boiler water treatment is necessary is addressed particularly to engineers concerned with the operation of smaller boilers and heating plants. The authors review—in a simple manner—the chemistry involved, and indicate briefly the effect of various impurities on boiler operation and maintenance. . . . A second article to appear in a later issue of HPAC will go further into the subject

*Ace Engineering Co.



Weighted average hardness, by states, of water furnished in 1932 by public supply systems in over 600 cities in the United States, including both surface and ground water supplies. Weighted average hardness of water supply in a state is secured by multiplying the average hardness of each supply by the number of consumers, and dividing the sum of these products by the number of consumers (from U. S. Geological Survey Water Supply Paper No. 658). The salts of calcium and magnesium make water hard

of the acid to form a *salt*. Calcium hydroxide and sulphuric acid, therefore, combine to make calcium sulphate, $CaSO_4$, and water.

Two salts can also undergo chemical reaction in which the acid radicals† interchange combination with the metals. Chemists call this reaction a "double decomposition," and express the change by means of an equation: $3CaSO_4 + 2FeCl_3 = 3CaCl_2 + Fe_2(SO_4)_3$. This equation does not necessarily signify that calcium sulphate and iron chloride always interact when in contact with one another, but merely states that should the conditions at the time of contact favor such chemical action, then three molecules of calcium sulphate will react with two molecules of iron chloride to form three molecules of calcium chloride and one molecule of ferric sulphate.

Concentration

Comprehension of water chemistry involves quantitative relations. Molecular relationships need not be discussed here, but it is necessary to explain the term *concentration*. By this expression is meant the quantity of a substance contained in a given quantity of water. In water chemistry, concentrations are expressed in either of two ways: chemists prefer to use parts per million (ppm) as the basis of analysis and calculation, while engineers are more familiar with quantities expressed as grains per gallon. A grain is 1/7000 of a pound, so it follows that one grain per gallon of water weighing 8⅓ pounds per gallon is 17.2 parts per million, and 1 ppm is 0.0582 grains per gallon.

Impurities in Water

The most important and most frequently encountered minerals in natural water supplies are salts of sodium, calcium and magnesium. Limestone rock consists of the mixed carbonates of calcium and magnesium, so it is easy to see why salts of these two metals are picked up by underground and surface waters. In general, well water contains higher concentrations of mineral matter than does water from surface streams or lakes. Sodium salts are so abundantly distributed in the earth that most waters contain some. Iron and aluminum are next in order of occurrence in waters, being usually found present as oxides. The most prevalent salts of calcium, magnesium, and sodium dissolved in water supplies are the bicarbonate, carbonate, and sulphate. Chlorides and nitrates would be next in frequency of occurrence, but are not so common as the first three named.

Effect of Hard Water

It is the salts of calcium and magnesium that make water *hard*. The term "hardness" is a reference to difficulty in making a lather with soap and the water in question. Calcium and magnesium combine with the soap to form insoluble compounds, and so additional soap must be supplied until the calcium and magnesium have been entirely converted into insolubles. Then only will soap be able to make suds or a lather. Hence, the usual

†The word "radical" as used by chemists signifies a group of atoms which act in the same manner as, and are replaceable by, a single atom, or a group of atoms remaining unchanged during a series of reactions, and so acting as the equivalent of a single atom.

method for testing waters for hardness is to add a measured quantity of a standard soap solution to a measured quantity of water sample. When sufficient soap solution has been added to create a permanent lather, the quantity of soap solution used is a measure of the water hardness.

While hardness as such is of vital importance to a laundry, because the consumption of soap for laundering is directly affected, the hardness of a boiler feedwater is merely an index of the calcium and magnesium salts present. In turn, this is an indication of the scale-forming qualities. In other words, when a hard water is fed into a boiler, scale will form on the surfaces of the tubes and plates, the amount and nature of the scale, and its deleterious effects upon the boiler, being dependent upon a number of factors.

Corrosion and Electrolysis

The carbonates and sulphates of calcium and magnesium are chiefly significant in boiler feed because of their scale-forming properties. Other salts, as chlorides and nitrates, may have other ill effects on the boiler. Under the influence of heat these salts react with water in a process called hydrolysis (the opposite of neutralization), in which certain acids are formed. These acids have a strongly corrosive action on boiler metal. Free oxygen dissolved in water also causes corrosion, especially if carbon dioxide is also present. Other substances contributing to corrosion of boilers are vegetable and animal oils and free acids. The former are saponified (converted into soap) by alkalies in the hot water with the consequent liberation of organic acids. Mineral acids are but seldom encountered in water supplies unless contaminated by industrial wastes, but when present are highly corrosive.

Electrolytic action can also occur within a boiler, under the influence of stray electric currents passing through the boiler or currents induced within the boiler by chemical or physical phenomena. The complete consideration of electrolytic effects is complex, and cannot be more than mentioned here. The effects of electrolysis resemble those of corrosion. Beyond exercising reasonable care in the installation and operation of all electric apparatus in the building, there is little that the plant operator can do to avoid electrolysis. Should conditions beyond his control, such as electric transmission or consumption in large amounts prevail on adjoining property, the introduction of a piece of zinc into the boiler will usually assist the situation.

Caustic Embrittlement

The failure of boilers due to caustic embrittlement can be prevented by correct water treatment. Caustic embrittlement refers to the process that gradually creates cracks between the crystals of the steel. A highly alkaline or caustic water induces caustic embrittlement, and correction of this water condition lies in maintaining a minimum ratio of alkali to sulphate.

Sodium salts do not contribute to hardness of water, and do not cause scale formation. Sodium hydroxide does induce embrittlement, as mentioned in the preceding paragraph, and sodium carbonate may decompose, under certain conditions, to form the hydroxide. Aside from the embrittlement factor, there is one other in boiler operation involving sodium compounds. A high concentration of sodium salts can cause the boiler water to prime or foam, with the result that liquid is carried out of the boiler with the steam. Usually, priming can be prevented by regulating the saline (salt) substance in the boiler water, largely through periodic blowdown.

Study Uses of Solar Heat

An experimental house designed to trap the sun's heat falling on the roof and store it in the basement for future use has been built for research on the utilization of solar radiation at Massachusetts Institute of Technology, Cambridge, Mass.

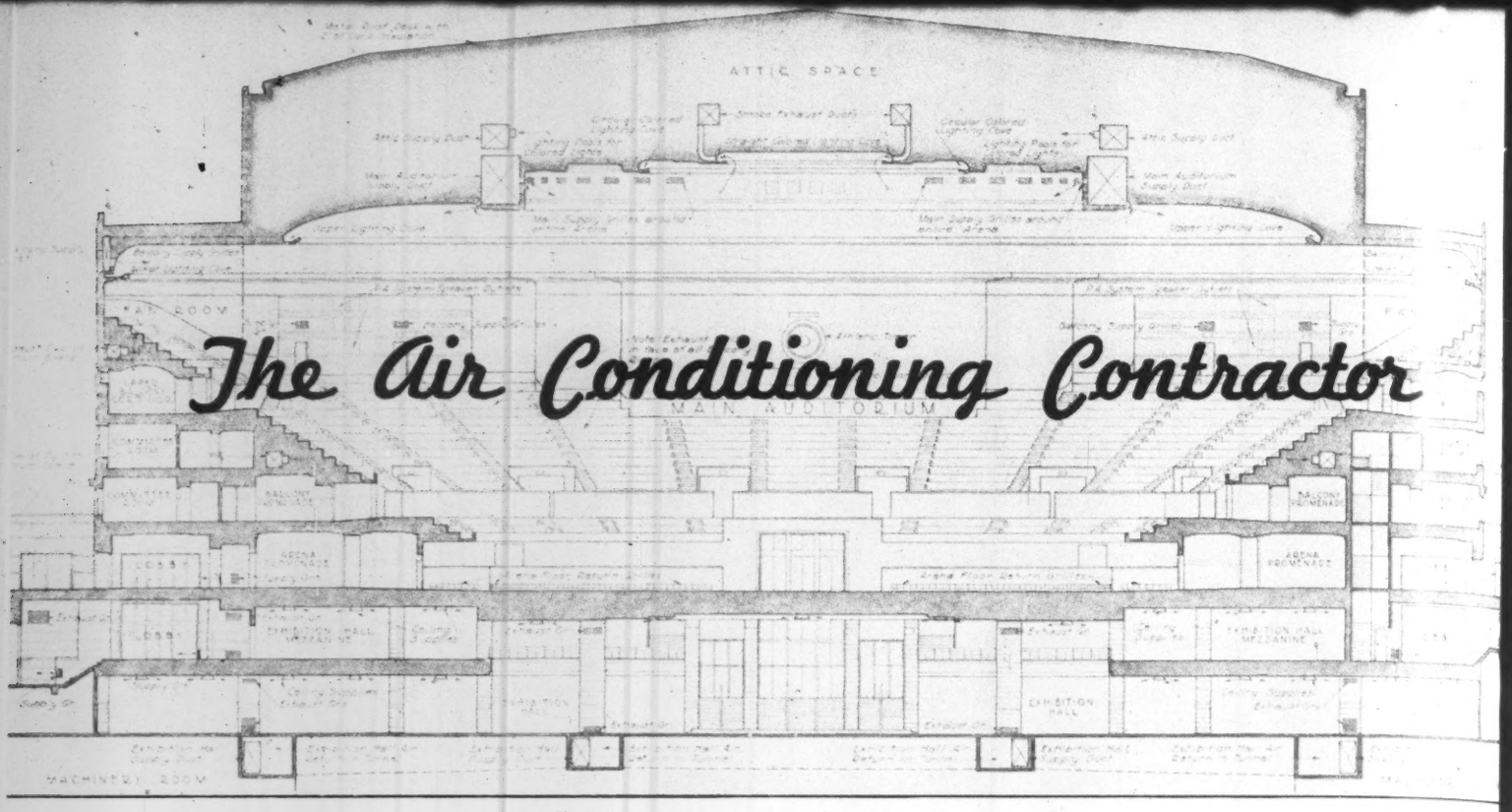
The purpose of the investigation is to study various uses of solar heat, including winter heating, summer air conditioning, and power generation. In the basement of the especially designed laboratory house is a large, well-insulated water storage tank which will be used for ironing out the fluctuations in heat collected from a source so variable as the sun. The building's heating system consists of a method of forced air circulation so arranged that the flow of air can be either over the hot tank surface or through the coils of a refrigeration system to be installed later. This refrigeration system, operating on an absorption principle, will utilize sunlight as its heat source.

While the MIT engineers are aware that the amount of solar heat in New England would make heating by solar radiation uneconomical in comparison with other sources of heat, there is sufficient sunshine in this region to test the efficiency of heating systems for localities where the climate is less rigorous.

The building in which the research is being conducted was designed especially for scientific studies and not as a model domestic dwelling. It is small, as buildings go, and has a large wall surface in proportion to its volume. Therefore an unusual amount of wall insulation was used to make the heating conditions comparable to a typical building of conventional size.

In conjunction with the research work on solar radiation, the institute has installed a pyrheliometer for recording the intensity of the sunlight throughout the day. This instrument makes it possible for MIT to coöperate with the U. S. Weather Bureau as a government subpyrheliometric station.

This research is one of several important projects planned under the solar energy program and made possible by a gift of nearly $650,000 presented by Dr. Godfrey L. Cabot of Boston for research on the utilization of solar radiation for the tasks of man. These studies are being carried on under the direction of a committee of which Professor Hoyt C. Hottel of the institute's department of chemical engineering is chairman. They will include fundamental investigations of direct physical and chemical methods of utilizing solar energy.



The Air Conditioning Contractor

Potential Volume, Overhead, Diversification and Employees' Functions

By John H. Carter*

THE chief attraction about the air conditioning business to the engineering type of mind is the fact that there is no sameness to it. Its applications are diverse, and it offers a challenge to the ingenuity and skill of the engineering contractor, which he will meet according to his own abilities. This diversity of problems, and the wide range of skill and adaptability to be found among contractors and engineers, make it a difficult task to outline a general plan of organization for an air conditioning contracting business, because the organization must reflect individuality—indeed, it must capitalize upon it.

The existing contracting organization should make the best of what it has as a nucleus, and should try to direct its growth or its alterations to adapt itself to its best field of action. The choice of this proper field lies outside the scope of this article, involving as it does many questions of local, special and personal import. This article can hope only to describe certain standardized problems of management, and relate how they have been solved in some cases, in the belief that the points described will clarify the solutions of somewhat parallel problems among other organizations.

The field of activity considered here is to consist mainly of medium and large installations of air conditioning, with possibly some secondary activity in small jobs handled by merchandising methods. The organization methods suggested will be based on the handling of larger sizes of complex engineered jobs, and would need some modification if they were to be applied to the small standardized work.

The first and fundamental point to consider is the potential of the territory, and the amount of dollar volume that the organization can reasonably expect to get as a yearly average. For the new organization there will of course be an introductory period during which volume is low, while the organization is learning the business and while the public is learning about the organization. The new organization, if its working capital is limited, will do well to take on one or more lines of equipment or supplies which do not require a long missionary period, in order to assure a certain amount of earnings while the representation of the major lines is being developed. A business in supplies and overhauls affords a remarkably useful backlog, and can be of the greatest help later, from the viewpoints of servicing and engineering experience and of sales assistance.

The estimated dollar volume will be based on coverage of a certain territory, whether or not the major line or lines are franchised with the contractor. The territory should be one which it will be economical for the contractor to cover. We can assume that the territory will be the usual wholesale trading area around the con-

*Kupferle-Hicks Heating Co.

A regular reader of HEATING, PIPING AND AIR CONDITIONING—an engineer with one of the oldest heating, ventilating and air conditioning contracting companies in the country—was recently given the job of re-organizing the entire outfit along modern lines. He wrote HPAC's editors for some information to assist him, and in his letter he said he'd like to see some articles on different phases of the subject in our pages. . . . We asked Mr. Carter to prepare discussions on some angles of the question, and the first of these is published here. It is planned to publish more of Mr. Carter's articles—and contributions from others—in future months. The editors will be interested to hear from any readers who would like specific points discussed, or with comments to offer

tractor's headquarters city. Coverage of a very wide area will probably not be practicable for the new or semi-experienced contracting organization. There are plenty of complications in contracting air conditioning right in the home town, and the big job in a distant city had better be left to the local contractors in that city, or to the highly experienced organizations which operate on a national scale.

The contractor had better do his own estimating of the potential volume which he can get, rather than take the possibly over-optimistic estimate of others. Special considerations enter the analysis. If the contractor will be able, through other business activities or through long-established reputation in an allied field, to get more than a newcomer's share of the market, he will be able to set up more elaborately than can the new contractor who lacks both public acceptance and ample working capital.

Whatever the estimate of dollar volume may be, and whoever makes it, it is the contractor who shoulders the overhead. He will set up his organization on a basis which bears a relation to the expected earnings, and he must consider that there will be fluctuations in the business cycle, no matter how carefully he diversifies his lines. The overhead in this kind of business is relatively inflexible, because the members of the organization are highly skilled along certain lines, and cannot be dropped and later replaced in the manner of department store extra salesgirls after Christmas. The contractor should proceed carefully, and base his hiring upon his own judgment, no matter what pressure may be put upon him. The author is acquainted with several successful air conditioning contractors, and they all have two things in common—they make their own decisions, and they have a stiff degree of sales resistance. Once the reasonable expectation of yearly dollar volume is estimated, the allowable overhead can then be determined.

A good organization will show a remarkable degree of flexibility. If the right key men are chosen, the organization can handle a volume twice as much as the normal, with the assistance of some extra draftsmen during the times when they are needed.

Since the overhead cannot very well be reduced materially once it is set up, and since there will be bad seasons now and then, it is desirable for the contractor to diversify both his lines handled and also the types of trade which he develops. Heating, ventilating, air conditioning and commercial refrigeration for industrial applications; industrial piping; power plant work, etc.—all offer a desirable diversification which will help to level out the cycle of business, where dependence upon residential and commercial work, for example, might result in unprofitable operations during occasional bad seasons. The field is less crowded in the more complex applications, and *if the organization is capable of handling them properly* it should be worthwhile developing this type of business.

The price quotation must of course include overhead as well as profit margin, and if too heavy an overhead must be charged the quotations are likely to be out of line. Overhead in the engineering-contracting field includes all items of expense, except sales commissions and job costs.

Once the overhead figure is established, the division of functions among the members of the organization can be set up—subject always to unclassifiable personal traits and abilities. Certain functions have to be covered, whether the organization is small or large. In the small organization one man may have to take care of the duties which would be covered by two or three men in a larger organization. Just which duties they are will depend upon the particular man and his experience and aptitudes. Spreading his attention among several specialties, he will probably not acquire the skill which a full-time man might have, but he will at least have the steadiness which comes from the assurance that his job is safer during a bad year. (This assurance of continuity will often carry weight with the customer, too.)

No matter how elaborate the organization may be, it is necessary that all members have a good understanding of the work (and some of the technique) of all of their colleagues. The nature of the contracting business is such that members of an organization often have to pinch-hit for each other, as for example during an emergency service call when the directly responsible fellow-employee is absent.

The functions which must be covered, whether the organization consists of two men or 20, are estimating, sales, layout, purchasing, installation and service, and management.

Considering the engineering-contracting field, the salesman is almost always an estimator and engineer. Sales ability, as such, is usually subordinated to engineering ability, since the nature of the work is so technical, and since the demand is usually already created in the customer's mind. The sales engineer who is capable of estimating his own job, and who can—and probably does—make a layout of the equipment and controls, will do a much cleaner job of selling, and his estimated cost is likely to hit the actual final cost very closely. In the small organization his remuneration will probably be a drawing account against a share of the net profits at the end of the year. In the large setup he will probably have a drawing account against a percentage of sales. If his remuneration in either organization is on a salary basis, he should participate in the fortunes of the firm by bonus or by salary adjustment.

Where there are several sales engineers, there will be a certain degree of specialization. If the lines handled show a seasonal curve of activity, it will be well to let each sales engineer represent two or more lines, so that he will be actively occupied the year around. While the men are to be considered specialists to some extent, they should nevertheless be able to handles sales for their associates, since many jobs call for several kinds of equipment on one order, and also since one specialist might be out traveling the territory, or working on a long-drawn-out distant job, when a rush estimate comes up.

[*A second article continuing this discussion of this subject will appear in an early issue.*]

Amount of Steam for Heating a Greenhouse Determined by Test

By G. K. Saurwein*

SCARCITY of accurate data on steam requirements for greenhouse heating prompted the installation of a condensate meter in the return line from the greenhouse situated on the roof of the biological laboratories at Harvard University, Cambridge, Mass. Fig. 1 shows the general arrangement of the greenhouse. Since the six rooms provide facilities for various types of plant growth and research work, different temperatures are maintained in them, varying from 44 to 72 F. Each room is heated by four horizontal pipe coils under the benches; each coil has about 240 lineal feet of 1½ in. pipe, and is controlled by a compressed air type modulating thermostat.

The meter was installed in October, 1936. The total condensate for the following 87 day period was 452,100 lb. Table 1 shows the amount of heat used, the average per day per degree difference between outside and inside temperatures per square foot of glass surface, and per cubic foot of space. The same table also shows the theoretical heat loss calculated by using coefficients from the ASHVE Guide, and assuming one air change per hour.

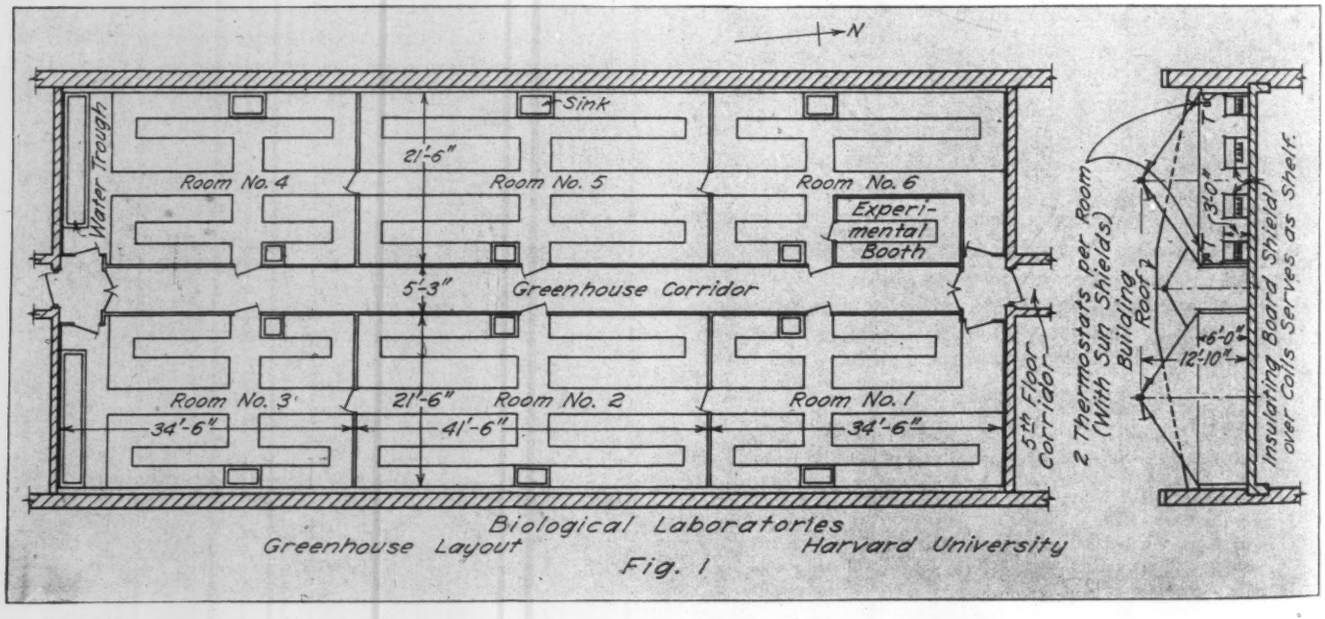
During the entire period under consideration the movable sash of the greenhouse glass was kept tightly closed. Moreover, the entrances at each end have air locks; that is to say, there are double doors 4 ft apart that serve as air locks and reduce the amount of air change. It is thus reasonable to assume one air change per hour. It was further assumed that the temperature at the 7 ft level near the center of each room was the average temperature for the entire room. Since the space below and at the ends of the greenhouse was heated, it was assumed that no heat transfer took place through the floor or end walls which adjoin heated space. It was further assumed that no heat was used to evaporate moisture. The theoretical heat loss checks reasonably with the measured heat supply, as can be seen from the table.

Table 1—Heat Used in Greenhouse

Steam condensate metered, lb	452,100
Heat in steam, Btu per lb	1,156
Heat in condensate, Btu per lb	100
Net heat supplied, Btu per lb	1,056
Total heat used, million Btu	477.4
Average outdoor temperature, F	38.4
Average inside temperature, F	63.8
Average difference between inside and outside temperatures, F	25.4
Number of days	87
Heat used per day, million Btu	5.49
Heat used per day per deg temp diff per sq ft of glass, Btu	35.0
Heat used per day per deg temp diff per cu ft of space, Btu	4.94
THEORETICAL CALCULATIONS	
Heat loss, Btu per hr per deg temp diff—	
Glass: 6,180 sq ft × 1.13[a]	6,990
Outside wall (9.7 in. brick): 1,324 sq ft × 0.37[a]	490
Corridor wall (6 in. brick): 670 sq ft × 0.50[a]	335
Volume (one air change): 43,740 cu ft × 0.018[b]	787
Total Btu per hr per deg temp diff	8,602
Theoretical Totals—	
Heat used per day, million Btu	5.24
Total heat used, million Btu	455.8
Heat used per day per deg temp diff per sq ft of glass, Btu	33.4
Heat used per day per deg temp diff per cu ft of space, Btu	4.72

*Superintendent, Engineering Service Dept., Harvard University.

[a]Coefficient of transmission (*U*) expressed in Btu per hr per sq ft per deg temperature difference.
[b]0.018 = Btu absorbed by 1 cu ft of air per deg.



Hangars at New York Airport Heated with Warm Air System

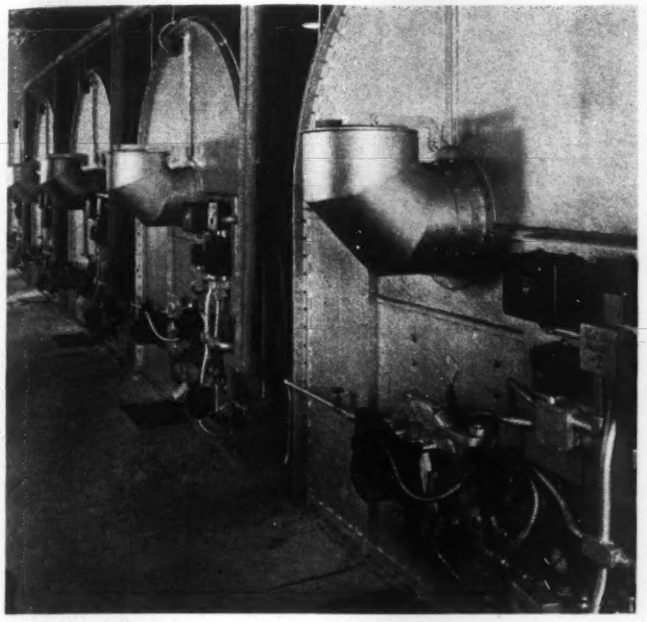


Fig. 1—Five of the heaters in the seaplane hangar heater room

For heating the seven hangars at New York's new municipal airport, a total of 30 oil fired direct air heaters are used, four each in the landplane hangars and six in the seaplane building. The heated air is distributed through underfloor ducts to continuous gratings in the floor just inside the doors, and to floor registers along the side walls. . . . Office spaces at the airport are heated by steam systems and cooled in summer by unit conditioners supplied with chilled water from a centrally situated refrigeration plant

NEW YORK CITY'S new municipal airport at North Beach, Queens, Long Island, comprises in its initial state seven large hangars—one for seaplanes and six for landplanes—and two administration buildings. The six landplane hangars are each 350 ft wide by 165 ft deep and 42 ft from ground to trusses, and have a volume of approximately 4,500,000 cu ft. The seaplane or marine hangar is a five sided building with doors on four sides and shop and office space in the rear, about 354 ft long and with a clearance to the roof trusses of 45 ft, and with a volume of about 6,000,000 cu ft.

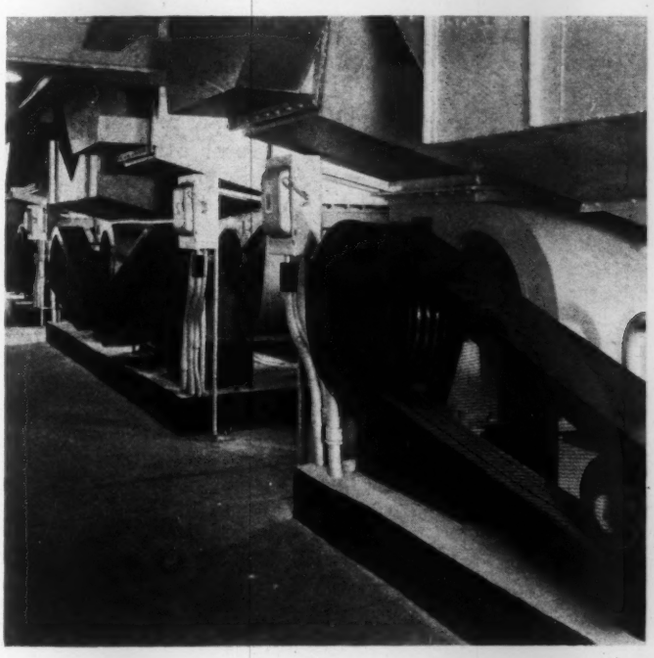
These hangars are heated by direct air heaters or furnaces of welded steel plate construction, fired with heavy No. 6 oil. Each of the heaters produces 3,260,000 Btu per hr; there are six heaters for the seaplane hangar and four for each of the six landplane hangars, or a total of 30 heaters with a combined capacity of 97,800,000 Btu. Each of the furnaces or heaters has two supply air blowers and an induced draft fan on a common shaft driven by a 25 hp motor. Each furnace, with its fans and burners, is a separate unit electrically interlocked with the necessary controls and safety devices to the adjoining units. The number of furnaces in operation at any time is automatically controlled according to the load.

An acceptance test of the six seaplane hangar units showed an efficiency of 86.68 per cent, CO_2 in exhaust gases of 13.42 per cent, exhaust gas temperature of 389 F and cfm of air heated per unit of 36,900.

Air is distributed in the hangars through underground masonry ducts to floor registers along the side walls and to a continuous grating about a foot wide in the floor just inside the large hangar doors. The heated air thus rises to form a curtain of warm air when the doors are open. Air is returned to the heaters through large grilles at floor level in the middle of the inside wall of the hangar. Heated air is distributed to the shop offices through metal duct work.

The hangar spaces are heated to 55 F and the adjoining shops and shop offices are heated to 65 and 70 F. The system is designed to raise the temperature in the hangar spaces from zero to 55 F in 24 min, after the doors are closed. Since the hangar doors have a clear

Fig. 2—Fan room or plenum chamber below the heaters



Most of the information given here is from a paper by Samuel E. Stott, Chief Engineer, Design Section, Works Progress Administration, which was presented at the annual meeting of the American Society of Mechanical Engineers and published in *Mechanical Engineering*, February, 1940, pp. 103-110. Illustrations and test results on heaters courtesy Dravo Corp.

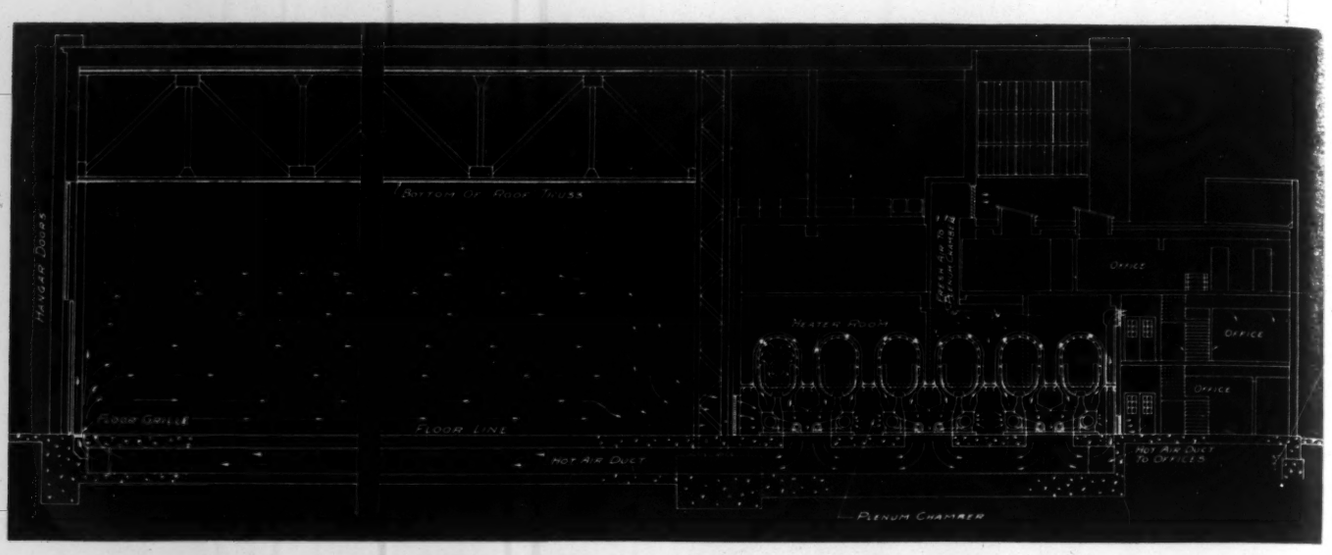


Fig. 3—How the seaplane hangar is heated

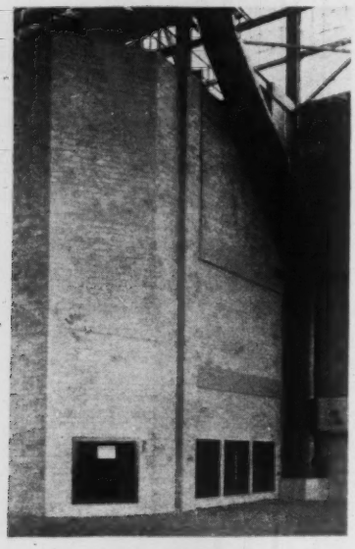


Fig. 4—Exterior of heater room, showing openings through which return air is drawn. The stack is for exhaust gases

opening of 330 ft width and 42 ft height, quick restoration of operating temperatures is important.

The two administration buildings are heated by conventional two-pipe, low pressure steam systems with vacuum return, under pneumatic control. Each building has its own low pressure steam plant, consisting of two welded steel boilers fired with No. 6 oil. In three of the hangars, the lean-to buildings are used as offices and are completely air conditioned. These spaces are heated by low pressure steam, and for summer conditioning, 22 unit conditioners are used. These units are supplied with chilled water from a central refrigerating plant containing four 50 ton "Freon" compressors, two water coolers, and four evaporative condensers. There is also a central refrigerating plant of 100 tons capacity serving non-office spaces that require air conditioning.

The planning division of the WPA, and Syska & Hennessy, consulting engineers, were responsible for the design of the airport. Delano & Aldrich were the architects. The airport has been named LaGuardia Field.

New Plant Heated by Steam and Hot Water

The heating for the Toledo Scale Co.'s new plant just outside Toledo, Ohio, is divided into two separate systems—an overhead steam unit heater system for the factory portion, and a forced hot water radiator system for the office section. The administration offices are embodied in the main building as a self-contained unit.

Insulated and covered underground steam mains run from the boiler house to the building, where direct connections are made for 100 lb steam for process use. Reducing valves supply 30 lb steam for heating.

The main portion of the factory is heated by projection type unit heaters. Direct horizontal suspended blower type unit heaters are situated at all large outside door openings. The projection type heaters, which are placed in the roof trusses, blow the air which is first drawn through grille type radiators, directly down onto the floor in such manner and at such velocity as to assure an even distribution of heat.

Each unit is controlled by a three-position magnetic starter situated in a convenient position, with "off," "hand" and "automatic" pushbuttons in the starter cover. The "automatic" position controls thermostats set for any desired temperature between 50 and 80 F. By placing in the "hand" position, it is possible to use the units for ventilation purposes in the summertime. Condensate is returned to a duplex pump and receiver set in the machine room, which discharges these returns direct to the boiler plant.

Radiators were required under the first floor windows of the office section, which meant that if steam were used there would have been either an underground return main or a return under the radiators along the wall. This being undesirable, a forced hot water system was selected; thus both the supply and return mains were installed on the first floor ceiling and in this manner the extra cost of underground returns and unsightliness of returns along the wall were eliminated.

Cast iron radiators were used throughout with extra coils installed in the skylights above the one-story portion of the plant. The hot water heat exchangers are controlled by means of thermostatic valves on the steam lines, which control the temperature of the water in relation to the outside temperature; that is, as the outside temperature rises, the water temperature from the heaters is lowered to the temperature required to maintain

[*Concluded on p.* 186]

Properties of Mixtures of Air and Saturated Water Vapor

By William Goodman*

Barometric Pressure 28 In. of Mercury

All tabulated values (except temperatures and pressures) are quantities per pound of dry air in the mixture.

Temp Deg Fahr t	Absolute Humidity: Pounds w	Absolute Humidity: Grains w	Volume: Dry Air Cu Ft v_D	Volume: Volume Increase of Humid Air Per Cent	Humid Specific Heat s'	$1061w$	Enthalpy (Heat Content): Water Vapor Btu h_V	Enthalpy (Heat Content): Air-Vapor Mixture Btu h
1	2	3	4	5	6	7	8	9
— 40	0.0001244	0.8705	11.30	0.01999	0.2401	0.1319	0.1297	—9.470
— 39	.0001316	0.9215	11.33	.02116	.2401	.1397	.1374	—9.223
— 38	.0001395	0.9763	11.36	.02242	.2401	.1480	.1456	—8.974
— 37	.0001477	1.034	11.39	.02374	.2401	.1567	.1542	—8.726
— 36	.0001563	1.094	11.41	.02513	.2401	.1658	.1633	—8.477
— 35	0.0001653	1.157	11.44	0.02657	0.2401	0.1753	0.1728	—8.287
— 34	.0001749	1.224	11.47	.02812	.2401	.1856	.1829	—7.977
— 33	.0001849	1.294	11.49	.02973	.2401	.1962	.1934	—7.727
— 32	.0001955	1.369	11.52	.03144	.2401	.2075	.2047	—7.475
— 31	.0002065	1.445	11.55	.03320	.2401	.2191	.2162	—7.224
— 30	0.0002183	1.528	11.57	0.03510	0.2401	0.2316	0.2287	—6.971
— 29	.0002306	1.614	11.60	.03707	.2401	.2446	.2416	—6.718
— 28	.0002433	1.703	11.63	.03911	.2401	.2581	.2550	—6.465
— 27	.0002569	1.798	11.66	.04130	.2401	.2726	.2695	—6.211
— 26	.0002710	1.897	11.68	.04357	.2401	.2875	.2844	—5.956
— 25	0.0002858	2.001	11.71	0.04595	0.2401	0.3033	0.3001	—5.700
— 24	.0003015	2.111	11.74	.04847	.2401	.3199	.3167	—5.443
— 23	.0003180	2.226	11.76	.05113	.2401	.3374	.3342	—5.186
— 22	.0003351	2.346	11.79	.05388	.2402	.3556	.3522	—4.928
— 21	.0003531	2.472	11.82	.05676	.2402	.3746	.3713	—4.669
— 20	0.0003720	2.604	11.84	0.05981	0.2402	0.3947	0.3914	—4.409
— 19	.0003920	2.744	11.87	.06302	.2402	.4159	.4125	—4.148
— 18	.0004125	2.888	11.90	.06632	.2402	.4377	.4343	—3.886
— 17	.0004342	3.039	11.92	.06980	.2402	.4606	.4573	—3.623
— 16	.0004572	3.200	11.95	.07350	.2402	.4850	.4818	—3.358
— 15	0.0004807	3.365	11.98	0.07728	0.2402	0.5100	0.5068	—3.093
— 14	.0005056	3.539	12.01	.08129	.2402	.5365	.5333	—2.827
— 13	.0005319	3.723	12.03	.08551	.2402	.5643	.5612	—2.559
— 12	.0005591	3.914	12.06	.08988	.2403	.5932	.5902	—2.290
— 11	.0005872	4.110	12.09	.09440	.2403	.6230	.6201	—2.020
— 10	0.0006175	4.323	12.11	0.09928	0.2403	0.6552	0.6524	—1.748
— 9	.0006484	4.539	12.14	.1042	.2403	.6879	.6853	—1.475
— 8	.0006814	4.770	12.17	.1096	.2403	.7230	.7205	—1.200
— 7	.0007150	5.005	12.19	.1149	.2403	.7586	.7564	—0.9236
— 6	.0007503	5.252	12.22	.1206	.2403	.7961	.7941	—0.6459
— 5	0.0007880	5.516	12.25	0.1267	0.2404	0.8360	0.8343	—0.3657
— 4	.0008265	5.786	12.27	.1329	.2404	0.8769	0.8755	—0.0846
— 3	.0008669	6.068	12.30	.1394	.2404	0.9198	0.9186	+0.1986
— 2	.0009095	6.367	12.33	.1462	.2404	0.9650	0.9642	0.4842
— 1	.0009540	6.678	12.36	.1534	.2404	1.012	1.012	0.7718
0	0.0009994	6.996	12.38	0.1607	0.2405	1.060	1.060	1.060
1	.001047	7.329	12.41	.1683	.2405	1.111	1.111	1.351
2	.001097	7.678	12.44	.1764	.2405	1.164	1.165	1.645
3	.001149	8.044	12.46	.1847	.2405	1.219	1.221	1.941
4	.001204	8.428	12.49	.1936	.2405	1.277	1.280	2.240
5	0.001260	8.822	12.52	0.2026	0.2406	1.337	1.340	2.540
6	.001319	9.232	12.54	.2120	.2406	1.399	1.403	2.843
7	.001380	9.662	12.57	.2219	.2406	1.465	1.469	3.149
8	.001444	10.11	12.60	.2322	.2407	1.532	1.538	3.458
9	.001512	10.58	12.62	.2430	.2407	1.604	1.610	3.770
10	0.001580	11.06	12.65	0.2541	0.2407	1.677	1.684	4.084
11	.001654	11.58	12.68	.2659	.2407	1.755	1.763	4.403
12	.001729	12.10	12.71	.2780	.2408	1.834	1.844	4.724
13	.001805	12.64	12.73	.2902	.2408	1.915	1.926	5.046
14	.001889	13.22	12.76	.3036	.2409	2.004	2.016	5.376
15	0.001973	13.81	12.79	0.3172	0.2409	2.094	2.107	5.707
16	.002061	14.42	12.81	.3313	.2409	2.186	2.201	6.041
17	.002152	15.06	12.84	.3459	.2410	2.283	2.300	6.380
18	.002247	15.73	12.87	.3612	.2410	2.384	2.402	6.722
19	.002346	16.42	12.89	.3772	.2411	2.489	2.510	7.070
20	0.002448	17.14	12.92	0.3935	0.2411	2.597	2.619	7.419
21	.002556	17.89	12.95	.4109	.2412	2.712	2.736	7.776
22	.002666	18.66	12.97	.4286	.2412	2.828	2.855	8.135
23	.002781	19.47	13.00	.4471	.2413	2.951	2.980	8.500
24	.002901	20.31	13.03	.4664	.2413	3.078	3.109	8.869
25	0.003023	21.16	13.06	0.4860	0.2414	3.208	3.242	9.242
26	.003154	22.08	13.08	.5071	.2414	3.346	3.383	9.623
27	.003286	23.00	13.11	.5283	.2415	3.487	3.527	10.01
28	.003424	23.97	13.14	.5505	.2415	3.633	3.676	10.40
29	.003568	24.98	13.16	.5737	.2416	3.786	3.833	10.79
30	0.003718	26.03	13.19	0.5978	0.2417	3.945	3.995	11.20
31	.003875	27.12	13.22	.6229	.2417	4.111	4.165	11.61
32	.004032	28.22	13.24	.6481	.2418	4.278	4.338	12.02
33	.004202	29.41	13.27	.6753	.2419	4.458	4.522	12.44
34	.004375	30.63	13.30	.7032	.2420	4.642	4.711	12.87
35	0.004554	31.88	13.32	0.7320	0.2421	4.832	4.905	13.31
36	.004742	33.19	13.35	.7621	.2421	5.031	5.109	13.75
37	.004936	34.55	13.38	.7930	.2422	5.237	5.321	14.20
38	.005135	35.95	13.41	.8254	.2423	5.449	5.538	14.66
39	.005342	37.40	13.43	.8584	.2424	5.668	5.764	15.12

Temp Deg Fahr t	Absolute Humidity: Pounds w	Absolute Humidity: Grains w	Volume: Dry Air Cu Ft v_D	Volume: Volume Increase of Humid Air Per Cent	Humid Specific Heat s'	$1061w$	Enthalpy (Heat Content): Water Vapor Btu h_V	Enthalpy (Heat Content): Air-Vapor Mixture Btu h
1	2	3	4	5	6	7	8	9
40	0.005557	38.90	13.46	0.8930	0.2425	5.896	5.997	15.60
41	.005778	40.45	13.49	0.9286	.2426	6.130	6.238	16.08
42	.006008	42.06	13.51	0.9654	.2427	6.375	6.490	16.57
43	.006245	43.71	13.54	1.004	.2428	6.626	6.748	17.07
44	.006491	45.44	13.57	1.043	.2429	6.887	7.017	17.58
45	0.006749	47.24	13.59	1.085	0.2430	7.161	7.299	18.10
46	.007013	49.09	13.62	1.127	.2432	7.441	7.587	18.63
47	.007285	51.00	13.65	1.171	.2433	7.729	7.885	19.17
48	.007569	52.98	13.68	1.216	.2434	8.030	8.195	19.72
49	.007861	55.03	13.70	1.263	.2435	8.341	8.515	20.28
50	0.008166	57.16	13.73	1.312	0.2437	8.664	8.850	20.85
51	.008482	59.38	13.76	1.363	.2438	9.000	9.196	21.44
52	.008802	61.61	13.78	1.415	.2440	9.339	9.545	22.03
53	.009142	64.00	13.81	1.469	.2441	9.700	9.919	22.64
54	.009485	66.39	13.84	1.524	.2443	10.06	10.30	23.26
55	0.009842	68.90	13.86	1.582	0.2444	10.44	10.69	23.89
56	.01022	71.51	13.89	1.641	.2446	10.84	11.10	24.54
57	.01060	74.17	13.92	1.702	.2448	11.24	11.51	25.19
58	.01099	76.95	13.94	1.766	.2450	11.66	11.95	25.87
59	.01140	79.79	13.97	1.831	.2451	12.09	12.40	26.56
60	0.01182	82.72	14.00	1.899	0.2453	12.54	12.86	27.26
61	.01225	85.78	14.03	1.969	.2455	13.00	13.34	27.98
62	.01271	88.98	14.05	2.041	.2457	13.49	13.84	28.72
63	.01318	92.24	14.08	2.116	.2459	13.98	14.35	29.47
64	.01365	95.55	14.11	2.193	.2461	14.48	14.87	30.23
65	0.01416	99.09	14.13	2.273	0.2464	15.02	15.43	31.03
66	.01466	102.6	14.16	2.355	.2466	15.55	15.99	31.83
67	.01519	106.4	14.19	2.440	.2468	16.12	16.58	32.66
68	.01574	110.2	14.21	2.528	.2471	16.70	17.18	33.50
69	.01630	114.1	14.24	2.618	.2473	17.30	17.80	34.36
70	0.01689	118.2	14.27	2.712	0.2476	17.92	18.44	35.24
71	.01748	122.4	14.29	2.808	.2479	18.55	19.11	36.15
72	.01811	126.8	14.32	2.908	.2482	19.21	19.80	37.08
73	.01875	131.2	14.35	3.011	.2484	19.89	20.50	38.02
74	.01941	135.9	14.38	3.117	.2487	20.59	21.24	39.00
75	0.02009	140.6	14.40	3.225	0.2490	21.31	21.99	39.99
76	.02079	145.5	14.43	3.338	.2494	22.06	22.76	41.00
77	.02153	150.7	14.46	3.456	.2497	22.84	23.58	42.06
78	.02227	155.9	14.48	3.576	.2500	23.63	24.41	43.13
79	.02304	161.3	14.51	3.700	.2504	24.45	25.26	44.22
80	0.02384	166.9	14.54	3.827	0.2507	25.29	26.14	45.34
81	.02467	172.7	14.56	3.960	.2511	26.17	27.06	46.50
82	.02551	178.6	14.59	4.095	.2515	27.07	28.00	47.68
83	.02639	184.7	14.62	4.236	.2519	28.00	28.97	48.89
84	.02728	191.0	14.64	4.380	.2523	28.95	29.97	50.13
85	0.02821	197.5	14.67	4.529	0.2527	29.93	31.00	51.40
86	.02918	204.2	14.70	4.683	.2531	30.96	32.07	52.71
87	.03017	211.2	14.73	4.841	.2536	32.01	33.18	54.06
88	.03118	218.3	14.75	5.005	.2540	33.08	34.30	55.42
89	.03224	225.7	14.78	5.175	.2545	34.21	35.49	56.85
90	0.03333	233.3	14.81	5.348	0.2550	35.36	36.69	58.29
91	.03445	241.2	14.83	5.528	.2555	36.55	37.94	59.78
92	.03559	249.2	14.86	5.712	.2560	37.77	39.22	61.30
93	.03679	257.5	14.89	5.903	.2566	39.03	40.55	62.87
94	.03802	266.2	14.91	6.100	.2571	40.34	41.92	64.48
95	0.03928	275.0	14.94	6.302	0.2577	41.68	43.33	66.13
96	.04058	284.1	14.97	6.511	.2583	43.06	44.78	67.82
97	.04192	293.4	14.99	6.725	.2589	44.47	46.27	69.55
98	.04332	303.2	15.02	6.949	.2595	45.96	47.84	71.36
99	.04475	313.2	15.05	7.179	.2601	47.48	49.44	73.20
100	0.04621	323.5	15.08	7.413	0.2608	49.03	51.07	75.07
101	.04773	334.1	15.10	7.657	.2615	50.64	52.77	77.01
102	.04930	345.1	15.13	7.907	.2622	52.31	54.53	79.01
103	.05092	356.5	15.16	8.165	.2630	54.03	56.35	81.07
104	.05259	368.1	15.18	8.434	.2637	55.80	58.21	83.17
105	0.05429	380.1	15.21	8.707	0.2644	57.61	60.12	85.32
106	.05607	392.5	15.24	8.992	.2652	59.49	62.11	87.55
107	.05790	405.3	15.26	9.284	.2661	61.44	64.17	89.85
108	.05978	418.5	15.29	9.586	.2669	63.43	66.27	92.19
109	.06173	432.1	15.32	9.896	.2678	65.49	68.45	94.61
110	0.06373	446.1	15.35	10.22	0.2687	67.62	70.71	97.11
111	.06578	460.5	15.37	10.55	.2696	69.80	73.01	99.65
112	.06792	475.5	15.40	10.89	.2706	72.06	75.41	102.29
113	.07013	490.9	15.43	11.24	.2716	74.41	77.89	105.01
114	.07240	506.8	15.45	11.60	.2726	76.81	80.44	107.80
115	0.07473	523.1	15.48	11.97	0.2736	79.29	83.07	110.67
116	.07716	540.1	15.51	12.36	.2747	81.87	85.80	113.64
117	.07965	557.6	15.53	12.76	.2758	84.51	88.61	116.69
118	.08221	575.5	15.56	13.17	.2770	87.23	91.48	119.80
119	.08484	593.9	15.59	13.60	.2782	90.02	94.44	123.00

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(Tabulations for 22 in., 23 in., 24 in., 25 in., 26 in. and 27 in. were published in July, August, September, October, November and December, 1939.)

Properties of Mixtures of Air and Saturated Water Vapor—(Continued)
Barometric Pressure 28 In. of Mercury

All tabulated values (except temperatures and pressures) are quantities per pound of dry air in the mixture.

Temp Deg Fahr	Absolute Humidity		Volume		Humid Specific Heat	1061w	Enthalpy (Heat Content)		Temp Deg Fahr	Absolute Humidity		Volume		Humid Specific Heat	1061w	Enthalpy (Heat Content)	
	Pounds	Grains	Dry Air Cu Ft	Volume Increase of Humid Air Per Cent			Water Vapor Btu	Air-Vapor Mixture Btu		Pounds	Grains	Dry Air Cu Ft	Volume Increase of Humid Air Per Cent			Water Vapor Btu	Air-Vapor Mixture Btu
t	w	w	v_D		s^*		h_V	h	t	w	w	v_D		s^*		h_V	h
1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
120	0.08760	613.2	15.61	14.04	0.2794	92.94	97.56	126.36	160	0.3295	2306.6	16.69	52.60	0.3883	349.62	372.42	410.82
121	.09043	633.0	15.64	14.48	.2807	95.95	100.75	129.79	161	.3418	2392.7	16.72	54.55	.3938	362.66	386.45	425.09
122	.09331	653.2	15.67	14.95	.2820	99.00	103.99	133.27	162	.3547	2483.2	16.75	56.61	.3996	376.38	401.21	440.09
123	.09634	674.4	15.70	15.43	.2834	102.21	107.40	136.92	163	.3682	2577.5	16.77	58.75	.4057	390.67	416.59	455.71
124	.09944	696.1	15.72	15.93	.2848	105.51	110.91	140.67	164	.3824	2676.9	16.80	61.02	.4121	405.74	432.81	472.17
125	0.1027	718.6	15.75	16.44	0.2862	108.92	114.55	144.55	165	0.3973	2780.9	16.83	63.38	0.4188	421.50	449.79	489.39
126	.1060	741.8	15.78	16.97	.2877	112.43	118.28	148.52	166	.4130	2890.9	16.85	65.88	.4258	438.17	467.74	507.58
127	.1094	765.9	15.80	17.52	.2892	116.08	122.17	152.65	167	.4294	3005.7	16.88	68.48	.4332	455.57	486.49	526.57
128	.1130	790.7	15.83	18.09	.2908	119.84	126.17	156.89	168	.4468	3127.7	16.91	71.25	.4411	474.07	506.41	546.73
129	.1166	816.3	15.86	18.68	.2925	123.72	130.30	161.26	169	.4650	3255.0	16.93	74.14	.4493	493.37	527.22	567.78
130	0.1205	843.3	15.88	19.28	0.2942	127.82	134.67	165.87	170	0.4843	3390.1	16.96	77.21	0.4579	513.84	549.29	590.09
131	.1244	870.6	15.91	19.90	.2960	131.96	139.08	170.52	171	.5046	3532.2	16.99	80.43	.4671	535.38	572.52	613.56
132	.1283	898.2	15.94	20.55	.2977	136.15	143.55	175.23	172	.5262	3683.3	17.02	83.86	.4768	558.29	597.23	638.51
133	.1325	927.8	15.96	21.21	.2996	140.62	148.33	180.25	173	.5491	3844.0	17.04	87.52	.4871	582.64	623.49	665.01
134	.1369	958.2	15.99	21.90	.3016	145.24	153.25	185.41	174	.5734	4013.7	17.07	91.36	.4980	608.37	651.26	693.02
135	0.1413	989.3	16.02	22.63	0.3036	149.95	158.28	190.68	175	0.5989	4192.1	17.10	95.40	0.5095	635.40	680.44	722.44
136	.1460	1021.7	16.05	23.36	.3057	154.85	163.51	196.15	176	.6262	4383.5	17.12	99.74	.5218	664.41	711.75	753.99
137	.1508	1055.4	16.07	24.12	.3079	159.97	168.98	201.86	177	.6554	4587.6	17.15	104.4	.5349	695.35	745.16	787.64
138	.1558	1090.7	16.10	24.92	.3101	165.31	174.69	207.81	178	.6864	4804.5	17.18	109.3	.5489	728.23	780.67	823.39
139	.1609	1126.4	16.13	25.74	.3124	170.73	180.48	213.84	179	.7197	5037.7	17.20	114.6	.5639	763.57	818.77	861.73
140	0.1662	1163.6	16.15	26.59	0.3148	176.37	186.51	220.11	180	0.7557	5289.6	17.23	120.3	0.5801	801.76	860.02	903.22
141	.1717	1201.8	16.18	27.47	.3173	182.16	192.71	226.55	181	.7939	5557.5	17.26	126.4	.5973	842.36	903.89	947.33
142	.1775	1242.7	16.21	28.38	.3199	188.36	199.33	233.41	182	.8355	5848.7	17.28	133.0	.6160	886.50	951.59	995.27
143	.1834	1283.5	16.23	29.33	.3225	194.55	205.95	240.27	183	.8805	6163.7	17.31	140.1	.6362	934.24	1003.2	1047.1
144	.1895	1326.5	16.26	30.31	.3253	201.06	212.92	247.48	184	.9289	6502.6	17.34	147.8	.6580	985.61	1058.7	1102.9
145	0.1959	1371.2	16.29	31.33	0.3282	207.84	220.20	255.00	185	0.9817	6871.9	17.37	156.2	0.6818	1041.6	1119.2	1163.6
146	.2026	1418.3	16.31	32.39	.3312	214.97	227.84	262.88	186	1.039	7270.2	17.39	165.2	.7074	1102.0	1184.5	1229.2
147	.2095	1466.5	16.34	33.48	.3343	222.28	235.67	270.95	187	1.101	7707.7	17.42	175.1	.7355	1168.3	1256.2	1301.1
148	.2167	1516.6	16.37	34.63	.3375	229.88	243.81	279.33	188	1.171	8194.9	17.45	186.1	.7668	1242.1	1336.1	1381.2
149	.2241	1568.4	16.40	35.81	.3408	237.73	252.22	287.98	189	1.247	8728.3	17.47	198.2	.8011	1323.0	1423.6	1469.0
150	0.2319	1623.1	16.42	37.05	0.3443	246.01	261.11	297.11	190	1.331	9318.4	17.50	211.6	0.8390	1412.4	1520.2	1565.8
151	.2399	1679.4	16.45	38.34	.3480	254.54	270.26	306.50	191	1.425	9977.8	17.53	226.5	0.8814	1512.3	1628.4	1674.2
152	.2480	1736.1	16.48	39.66	.3516	263.14	279.48	315.96	192	1.531	10,716	17.55	243.3	0.9289	1624.2	1749.4	1795.5
153	.2570	1799.1	16.50	41.06	.3557	272.70	289.74	326.46	193	1.650	11,553	17.58	262.2	0.9827	1751.1	1886.7	1933.1
154	.2661	1862.6	16.53	42.50	.3597	282.32	300.07	337.03	194	1.786	12,501	17.61	283.7	1.044	1894.8	2042.4	2088.9
155	0.2756	1929.3	16.56	44.02	0.3640	292.43	310.93	348.13	195	1.942	13,592	17.63	308.4	1.114	2060.1	2221.3	2268.1
156	.2855	1998.4	16.58	45.59	.3685	302.90	322.20	359.64	196	2.122	14,855	17.66	337.0	1.195	2251.5	2428.5	2475.6
157	.2956	2069.2	16.61	47.22	.3730	313.63	333.73	371.41	197	2.334	16,340	17.69	370.6	1.290	2476.7	2672.1	2719.4
158	.3065	2145.6	16.64	48.94	.3779	325.21	346.17	384.09	198	2.586	18,102	17.72	410.5	1.404	2743.7	2961.2	3008.8
159	.3177	2224.1	16.66	50.72	.3830	337.11	358.97	397.13	199	2.823	19,762	17.74	448.1	1.510	2995.4	3234.0	3281.7
									200	3.263	22,839	17.77	517.7	1.708	3461.7	3738.7	3786.7

[*Concluded from p.* 184]

the desired temperature in the offices (approximately 72 F).

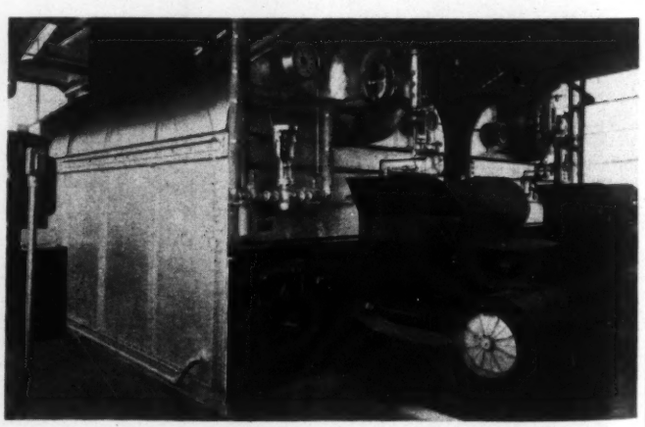
The control system consists of an outdoor thermostat, an indoor limiting thermostat, motorized valves, relays, switches, etc. The control device meters the necessary amount of heat into the building to offset the heat losses caused by outside resultant temperatures. Resultant temperatures can be summed up as the composite of factors that affect the rate of heat loss: outside temperature, solar radiation, wind direction and wind velocity.

The control comprises a block of metal, an electric heater, and a thermostat, in a small housing. Its designed temperature is adjustable and can be varied to suit the building. As the heat source of the building and the electric current to the control heater are operated simultaneously, both at the command of the thermostat, the correct amount of heat to the building to offset its heat losses can be metered.

Albert Kahn, Inc., were the architects and engineers, and furnished the above information on this plant.

Unit heaters are mounted in the trusses

The boiler house supplies heating and process steam



Heating · Piping and Air Conditioning

AMERICAN SOCIETY of HEATING and VENTILATING ENGINEERS

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SOUTHERN CALIFORNIA: *Organized,* 1930. *Headquarters,* Los Angeles, Calif. *Meets,* Second Tuesday in Month. *President,* J. F. PARK, 1234 S. Grand. *Secretary,* E. P. WELLS, 2730 E. Eleventh St.

WASHINGTON, D. C.: *Organized,* 1935. *Headquarters,* Washington, D. C. *Meets,* Second Wednesday in Month. *President,* T. H. URDAHL, 726 Jackson Pl., N. W. *Secretary,* P. H. LOUGHRAN, JR., 4513—49th St.

WESTERN MICHIGAN: *Organized,* 1931. *Headquarters,* Grand Rapids, Mich. *Meets,* Second Monday in Month. *President,* B. F. McLOUTH, 135 Gunson St., E. Lansing, Mich. *Secretary,* C. H. PESTERFIELD, Mich. State College, E. Lansing, Mich.

WESTERN NEW YORK: *Organized,* 1919. *Headquarters,* Buffalo, N. Y. *Meets,* Second Monday in Month. *President,* L. P. SAUNDERS, 507 Pine St., Lockport, N. Y. *Secretary,* H. C. SCHAFER, 45 Church St., Buffalo, N. Y.

WISCONSIN: *Organized,* 1922. *Headquarters,* Milwaukee, Wis. *Meets,* Third Monday. *President,* H. C. FRENTZEL, 4363 N. Wildwood Ave. *Secretary,* W. A. OUWENEEL, 801 Marshall Ave., South Milwaukee, Wis.

Diminishing Effectiveness of Successive Thicknesses of Insulating Materials

By Paul D. Close*, (*MEMBER*), Chicago, Ill.

PRACTICALLY everyone is familiar, consciously or otherwise, with examples in everyday experience of the principle of diminishing return. In the automotive field, for example, it is generally known that the speed of an automobile or an airplane does not increase in proportion to the horsepower developed. Because of wind resistance and other factors, each additional horsepower developed will not increase the speed of a vehicle the same as the preceding horsepower. With the average passenger car, the horsepower and fuel required to increase the speed beyond say 50 or 60 mph increases very rapidly until a practical limit is reached.

So it is with thermal insulating materials. Heating engineers, of course, are familiar with the effect of diminishing return on the value of insulating materials. However, those who have not had occasion thoroughly to delve into the subject may not be aware of the magnitude or extent of the effect of this principle as applied to everyday heating and insulation problems.

Diminishing return is the result of an approach toward some limit which is never reached except at infinity. In the case of insulating materials, the limit is a structure or barrier having a zero rate of heat loss which theoretically is attained by means of an infinite thickness of insulation, regardless of its efficiency. As this limit is approached, the effect of each successive increment of insulation thickness diminishes and finally reaches a point which may be regarded as the practical limit when considered from the standpoint of its effectiveness. This paper is intended solely to illustrate the diminishing effectiveness of successive thicknesses of insulating materials irrespective of economic[1] considerations which would necessarily involve heating and insulation costs, depreciation, and other variables entering into such problems.

The diminishing effect of various thicknesses of insulation applied to walls and roofs of various heat loss coefficients is shown in Fig. 1. The left hand vertical scale (ordinate) indicates the heat loss coefficients of the wall or roof, whereas the horizontal scale (abscissa) shows thicknesses of insulation up to 5 in. Curves are shown for walls or roofs having coefficients up to 0.80 and the coefficients of these constructions with various thicknesses of insulation may be determined by noting the intersections of the curves with the insulation thickness under consideration and referring to the left hand scale. For example, the coefficient of a 0.50 roof (without insulation) is 0.20 with 1 in. of insulation. The curves of Fig. 1 apply also to walls for similar conditions and the positions of several common walls and roofs are indicated on the left hand scale.

Initial Thicknesses Most Effective

It is apparent from the curves that the initial thicknesses of insulation are by far the most effective in reducing the rate of heat flow through the wall or roof, and that from 1 in. to 2 in. of insulation will reduce the heat flow adequately in the majority of cases, depending upon the initial coefficient. The curves become asymptotic or flatten out at from about 1 in. to 2 in. of insulation—about 1 in. for a 0.20 wall or roof to about 2 in. for a 0.80 wall or roof. This is not necessarily the economic thickness[1] nor does it mean that greater thicknesses may not be warranted in certain cases, but it does mean that the most effective results are obtained by the first inch or two of insulation.

Another interesting basis of comparison is the percentage reduction in the rate of heat flow. Percentages for walls and roofs of various coefficients up to 0.80 and for thicknesses of insulation up to 4 in. are shown in Fig. 2 and Table 1. The percentages shown may be regarded as the percentages of heat or fuel saved, considering the wall or roof only, for the thicknesses of insulation involved.

Relative Values

The relative values of various thicknesses of insulation are readily apparent from the curves of Fig. 2, or from Table 1. If the wall or roof coefficient is 0.30 (without insulation), the first ½ in. of insulation will reduce the heat loss through the wall or roof to the extent of 31.3 per cent, the second ½ in. an additional 16.2 per cent, the third ½ in. an additional 10.2 per cent, and the fourth ½ in. an additional 6.7 per cent.

The first 1 in. of insulation, which will reduce the heat loss through a 0.30 wall or roof 47.5 per cent, is more effective than the next 3 in. of insulation which reduce the rate of heat loss an additional 33.0 per cent. This next 3 in. of insulation are only 69.5 per cent as

[1]The economic thickness is usually taken to be the thickness which will produce the minimum combined annual cost of insulation, based on the proper annual fixed charges, and the annual cost of heat lost through the wall. See, When Does Insulation Pay? by Paul D. Close. (*Heating, Piping and Air Conditioning*, December, 1934).

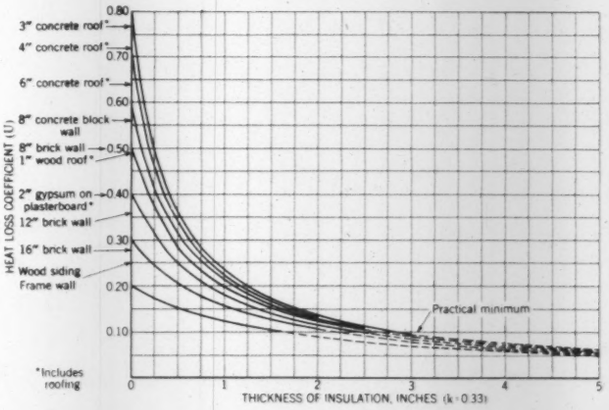

Fig. 1—Curves showing heat loss coefficients for various thicknesses of insulation

*Technical Secretary, Insulation Board Institute.
Presented at the 46th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Cleveland, Ohio, January, 1940.

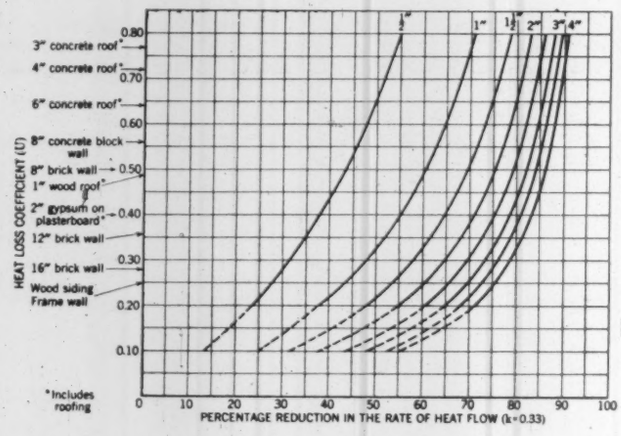

Fig. 2—Curves showing percentage reduction in the rate of heat flow plotted against wall or roof coefficient

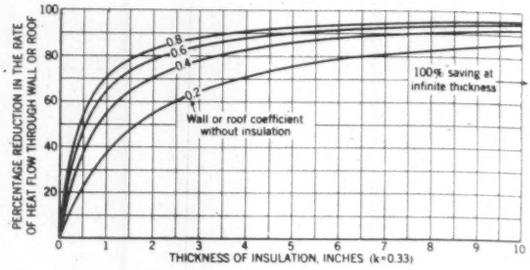
PERCENTAGE REDUCTION IN THE RATE OF HEAT FLOW THROUGH WALL OR ROOF
THICKNESS OF INSULATION, INCHES (k=0.33)
100% saving at infinite thickness
Wall or roof coefficient without insulation

Fig. 3—Curves showing percentage reduction in the rate of heat flow plotted against insulation thickness

effective as the first 1 in. of insulation in this case. Because of the rapidly diminishing effect of each inch of insulation it would actually require nearly 20 in. of insulation in this case to double the effect of the first inch of insulation, or the first inch of insulation is equal in insulating effect to the next 20 in. For walls or roofs of other coefficients, the actual ratios and percentages are of course somewhat different, but the general principle is the same, i.e. that the first inch of insulation is by far the most effective or valuable and each additional inch is less and less effective or valuable from the standpoint of heat or fuel saving.

The percentage reduction in the rate of heat flow through the wall or roof plotted against thickness of insulation, for various wall or roof coefficients is shown in Fig. 3. These curves are based on Table 1 except that the insulation thickness is extended to 10 in. An infinite thickness would be required to reduce the heat flow 100 per cent.

The question may arise as to the net effect of materials which replace others in the building construction, as for example, structural insulating board sheathing used in place of wood sheathing, and structural insulating board lath used in place of wood lath. The net effect of insulating board sheathing $\frac{25}{32}$-in. thick, used in place of wood sheathing is equivalent to a net thickness of slightly more than ½ in. of insulation.[2] The value of ½-in. insulating board lath used in place of wood lath is equivalent to slightly less than ½ in. of insulating board.[3] The combined net value of the insulating board sheathing and lath is slightly less than 1 in.[4] where the ½-in. insulating board lath is used, and slightly less than 1½ in. (1.42 in.) where the 1-in. lath is used in combination with insulating board sheathing. This 1 in. or 1½ in. of insulation are not only the most valuable or effective, and accomplish considerably more than the next several inches of insulation which may be installed, but they are also the least costly in most cases because other structural materials are replaced. This is especially true of insulating board sheathing, which usually costs little or no more than the wood sheathing replaced.

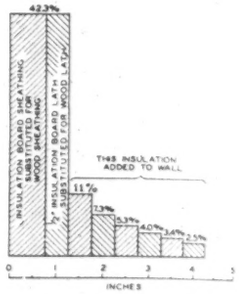

Fig. 4 — Chart showing diminishing efficiencies of insulation

In order to illustrate the net value of the insulation when used in place of other materials, consider the average or composite frame wall with exterior finishes of siding, shingles, brick veneer and stucco. If $\frac{25}{32}$-in. insulating sheathing and ½-in. lath are used in place of wood sheathing and lath, the heat loss through the wall will be reduced 42.3 per cent.[5] The next ½ in. would reduce the heat loss, or would save, an additional 11 per cent. Therefore, if insulating board sheathing and 1-in. insulating board lath are used the saving in the heat loss through the walls would be 53.3 per cent. The next ½ in. of insulation would save 7.3 per cent, the next 5.3 per cent, the next 4.0 per cent, the next 3.4 per cent and the next 2.5 per cent, as shown in Fig. 4.

Table 1—Percentage Savings Due to Various Thicknesses of Insulation Applied to Wall or Roof[a]

Coefficient	Insulation Thickness (Inches)							
	½	1	1½	2	2½	3	3½	4
	Per cent							
0.80	54.8	70.8	78.5	82.8	85.9	87.9	89.4	90.7
0.70	51.6	68.0	76.1	80.9	84.1	86.4	88.1	89.5
0.60	47.7	64.5	73.2	78.3	82.0	84.5	86.4	87.9
0.50	43.2	60.4	69.4	75.2	79.1	82.0	84.1	85.84
0.40	37.5	55.0	64.5	70.75	75.0	78.4	80.9	82.9
0.30	31.3	47.5	57.7	64.4	69.4	73.2	76.1	78.4
0.20	23.5	37.75	47.75	54.75	60.25	64.5	68.0	70.74

[a] Conductivity k assumed to be 0.33. These percentages represent the reduction in the rate of heat flow and apply to the net wall or roof area insulated and not to the building as a whole. Figures are based on the insulation being *added* to the structure, and not replacing other materials or filling air space.

[2] Resistance of 25/32-in. insulating board sheathing, 2.37; resistance of wood, 0.82; net resistance, 2.37 — 0.82 = 1.55 or the equivalent of slightly more than ½ in. of insulating board.

[3] Resistances of insulating board lath and plaster, 1.665; resistance of wood lath and plaster, 0.40; net resistance 1.265.

[4] $\frac{1.265 + 1.55}{3.03} = 0.93$ in.

[5] The average coefficient of an uninsulated frame wall is 0.27. If 25/32-in. insulation board sheathing and ½-in. lath are used in place of other materials, the average coefficient will be reduced to 0.154, based on a conductivity of 0.33, or 42.3 per cent.

Application of Data to Building

Thus far the walls and roof have been considered individually rather than in their relation to the total heat losses of the building. The next question then is, how do those principles apply when the entire structure is under consideration?

In order to throw some light on this subject, an analysis was made of an actual American Colonial residence, Fig. 5 containing 7 rooms, 2½ baths and a base-

ment recreation room, located in Kenilworth, Ill. The walls are of brick veneer frame construction and the roof consists of asphalt shingles on wood sheathing. The cubage is approximately 27,500 cu ft.

The calculations were based on the methods and constants set forth in the ASHVE GUIDE. The inside and outside design temperatures were 70 F and minus 10 F respectively and the wind velocity was assumed to be 12.5 mph. The heat losses for the uninsulated building are tabulated by rooms in Table 2. The percentage losses are shown for the various surfaces and are 33.7 per cent for walls, 20.6 per cent for the ceiling and roof, 3.1 per cent for the floors, 28.6 per cent for the glass and door losses and 14.0 per cent for the losses due to air leakage or infiltration.

The total heat loss of 109,790 Btu for the uninsulated building can, of course, be reduced by the use of various materials such as weatherstrips, storm windows and wall and roof insulation. The problem then is to determine what material or combination of materials will give the most satisfactory results from the standpoint of cost, balanced heat losses and avoidance of possible dangers, such as moisture troubles. It might be possible to use say 20 in. of insulation in the walls and to neglect the roof or the windows or the air leakage. But this 20 in. of insulation would be very costly and would be only twice as effective approximately, because of diminishing return as about 1 in. of insulation. There would still be an excessive loss through the other members of the structure. Thus instead of using 20 in. of insulation in the walls, the results would be far more satisfactory to use approximately 1 or 2 in. of insulation at this location and to apply the difference in the cost of the 20 in. of insulation, or part of it, toward reducing the heat losses through the roof and glass areas.

If weatherstrips are used, the infiltration losses will be reduced from 15,380 to 10,970, or 4,410 Btu per hour, a saving of 4.2 per cent. (See Fig. 6.) If storm sash are used in combination with weatherstrips[6] on all above-grade windows except the garage, the heat loss will be reduced 17,900 Btu per hour, or 16.3 per cent.

Structural insulating board used throughout will reduce the heat loss 31.1 per cent. In this case, $\frac{2}{3}\frac{5}{2}$-in. insulating board sheathing is used in place of wood sheathing and ½-in. insulating board lath and plaster is used in the walls and in the garage ceiling in place of metal lath and plaster; also 1-in. insulating board

[6]Storm sash reduce the loss through the glass areas and also serve the same purpose as weatherstrips; in fact, it is customary to assume the same leakage for certain windows with storm sash as with the same windows without storm sash but with weatherstrips.

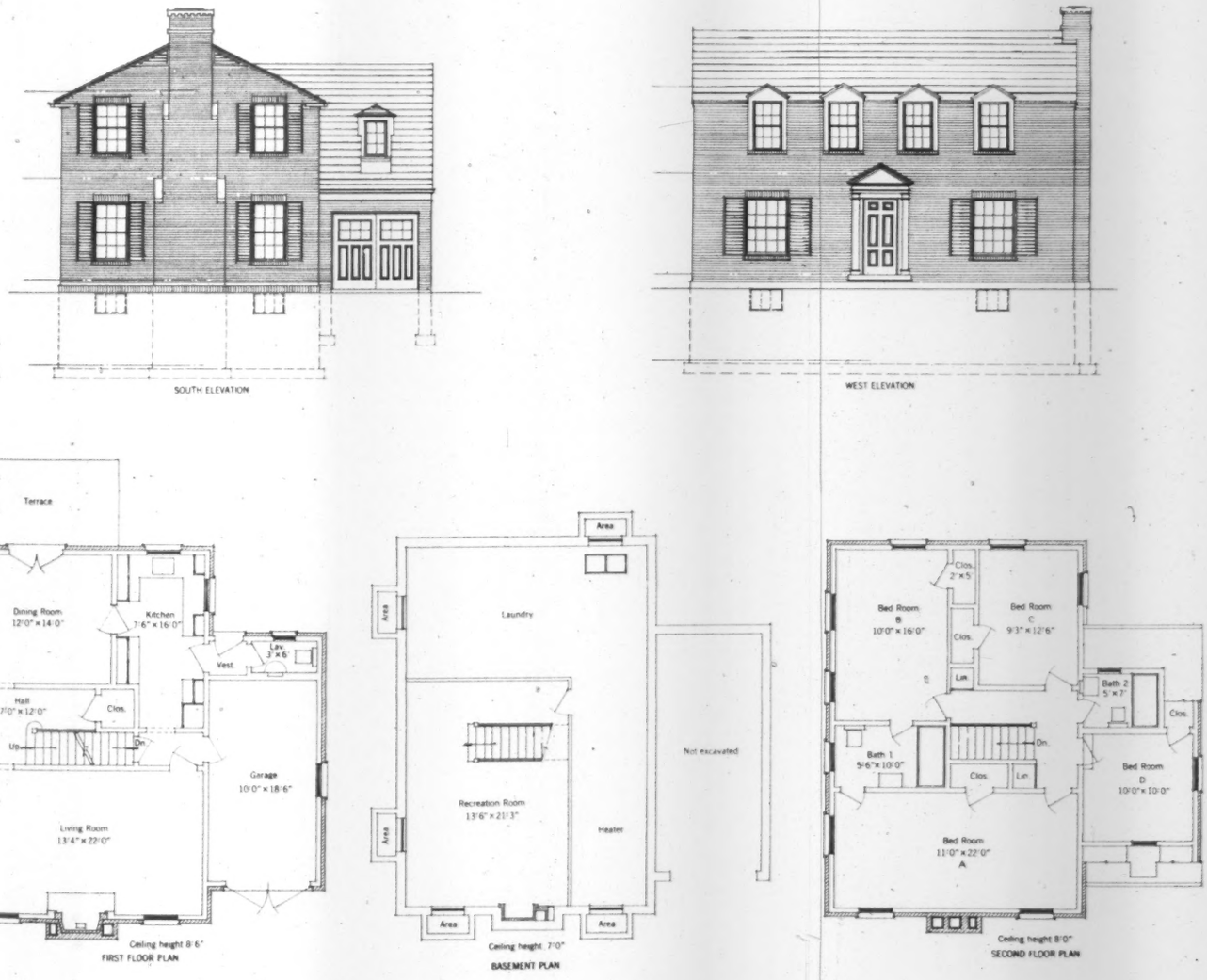

Fig. 5—South and west elevations and floor plans of residence

Table 2—Summary of Heat Losses for Uninsulated Building (*No Insulation, Weatherstrips or Storm Windows*)

Room or Space	Temperature Maintained	Walls	Ceiling and Roof	Floor	Glass and Door (D=door)	Infiltration	Total (Btu per Hour)
Bedroom *A*	70	5,330	6,660	.....	3,620	1,580	17,190
Bedroom *B*	70	3,490	4,400	.....	3,620	1,580	13,090
Bedroom *C*	70	2,560	3,300	.....	2,460	790	9,110
Bedroom *D*	70	2,650	3,300	960[a]	1,810	790	9,510
Bathroom *1*	70	670	1,510	.....	1,270	790	4,240
Bathroom *2*	70	1,770	960	310[a]	810	660	4,510
Living Room	70	5,980 1,280[a]	.....	.....	4,520	1,750	13,530
Dining Room	70	3,720	.....	.....	4,980	1,360	10,060
Kitchen	70	2,150 700[a]	.....	.....	1,630 (D) 300[a]	1,180	5,960
Lavette	70	1,840 1,160[a]	.....	.....	810 (D) 780	830	5,420
Entrance Hall	70	870	2,490	.....	640	880	4,880
Garage	35	2,110 -3,140[a]	.....	-1,180[b] -1,270[a]	2,700 (D) 300[a] (D)1,010	2,700	2,630
Recreation Room	70	3,850	.....	4,600	720	490	9,660
Totals		36,990	22,620	3,420	31,380	15,380	109,790
Percentages		33.7	20.6	3.1	28.6	14.0	100.0

[a] The heat losses into the garage from adjoining rooms are heat gains for the garage and are designated as minus (−) heat losses. For example, the floor heat losses of Bedroom *D* (960) and Bathroom *2* (310) represent a heat gain through the ceiling of the garage of 1270. As the quantities cancel they are shown in the same column (Floor) rather than in the floor column for Bedroom *D* and Bathroom *2*, and the ceiling column for the garage.

[b] As the garage temperature is assumed to be 35 F and the ground temperature is 45 F there is a heat gain through the garage floor, or a heat loss of −1,180.

lath and plaster in the second floor ceiling in place of metal lath and plaster, ½-in. insulating board on top of ceiling joists, 1-in. insulating board in the recreation room floor and ½-in interior finish on the recreation room walls. If 1-in. insulating board lath is used in the outside walls instead of ½-in. there will be an additional saving of 2.3 per cent, the total saving being 33.4 per cent. The maximum saving due to the installation of 3⅝ in. of insulation in both the walls and the top floor ceiling is 37.7 per cent[7] which is only 4.2 per cent more than with structural insulating board. Thus, considering the building as a whole, the difference is negligible due primarily to the law of diminishing return and is by no means in proportion to the relative thicknesses of the materials.

But insulating board in combination with storm sash and weatherstrips will reduce the heat loss in this case nearly 50 per cent, or 49.7 per cent, to be exact. Thus instead of installing the maximum possible thickness of insulation in the walls and roof and neglecting the windows, and thereby endeavoring to reduce these losses to the practical minimum against the resistance of the law of diminishing return, more satisfactory results can be obtained by distributing the *insulation*. Using insulating board in combination with storm windows and weatherstrips results in a greater combined saving and provides a more balanced heat loss.

Conclusions

1. Because of diminishing return, the first increment of thickness of insulation is the most effective and each successive increment of thickness is less and less effective.

[7] This is based on a theoretical wall coefficient of 0.074 and a ceiling coefficient of 0.079. However, many engineers for practical reasons, use minimum coefficients of 0.10, even though the theoretical coefficients may be below this value. Therefore, the practical saving would be somewhat less, based on a coefficient of 0.10.

Table 3—Summary of Heat Losses for Building with Weatherstrips, Storm Sash and Insulation

No.	Construction and Insulation	Heat Losses (Btu per Hour) Walls	Ceiling and Roof	Floor	Glass and Door	Infiltration	Totals	Per Cent Saving Based on No. 1
1	Walls: Brick Veneer, wood sheathing, studs, metal lath and plaster. Roof: asphalt shingles on wood sheathing: ceiling, metal lath and plaster. No insulation, weatherstrips or storm sash	36,990	22,620	3,420	31,380	15,380	109,790	
2	Same as 1, but with weatherstrips (Except garage doors)	36,990	22,620	3,420	31,380	10,970	105,380	4.2
3	Same as 2, but with storm sash	36,990	22,620	3,420	17,890	10,970	91,890	16.4
4	Insulating board sheathing ½ in. ins. bd. lath on walls and garage ceiling; 1 in. ins. bd. lath on 2nd fl. ceiling; 1 in. ins. bd. underrecreation room floor; ½ in. ins. bd. on recreation room walls; storm sash and weatherstrips	20,430	8,010	400	17,890	10,970	57,700	47.4
5	Same as 4 but with 1 in. ins. bd. lath in walls instead of ½ in.	17,950	8,010	400	17,890	10,970	55,220	49.7

2. The rate of diminishing return, however, depends on the initial wall or roof coefficient and the point at which the curve begins to flatten out or become asymptotic ranges from about 1 in. of insulation for a 0.20 wall or roof to about 2 in. of insulation for a 0.80 wall or roof.

3. Thus this range of thickness—1 in. to 2 in. or an average of 1½ in.—may be regarded as the limit of maximum effectiveness, beyond which the effective value of the insulation is extremely limited.

4. The first 1-in. of insulation added to a 0.30 wall or roof will reduce the heat loss through the wall or roof to the extent of 47.5 per cent for the conditions under consideration. The next 3 in. will reduce the heat loss an additional 33.0 per cent. Therefore, this next 3 in. are only 69.5 per cent as effective as the first 1 in. of insulation. Because of the rapidly diminishing return it would actually require 20 in. of insulation in this case to produce the same results as the first 1 in. of insulation.

5. When wood sheathing and lath are replaced with 25/32-in. insulating board sheathing and ½-in. insulating board lath, the heat loss through the wall is reduced 42.3 per cent. If 1-in. lath is used in place of ½-in. there would be an additional saving considering the walls only, of 11.0 per cent and the combined saving would be 53.3 per cent. The next ½ in. of insulation would save 7.3 per cent, the next 5.3 per cent, the next 4.0 per cent, the next 3.4 per cent, the next 2.5 per cent.

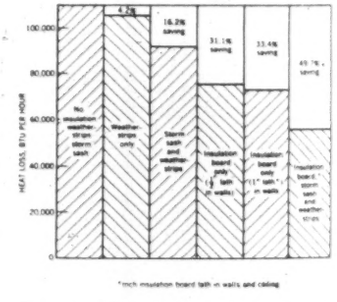

Fig. 6—Heat losses and percentage savings due to various materials

6. Similar results were obtained when the law of diminishing return was applied to the total heat losses of a residence. Insulating board used throughout showed a saving of 33.4 per cent whereas the maximum saving obtained by the use of wall-thick materials was only 4.2 per cent greater, or 37.7 per cent. By distributing the insulation over all wall, roof and glass surfaces, using insulating board in combination with storm windows and weatherstrips, instead of concentrating the insulation in the wall and roof areas, the total saving was increased to nearly 50 per cent, as compared to the uninsulated building.

Calculation of Coil Surface Areas for Air Cooling and Dehumidification

By John McElgin* and D. C. Wiley,** (*MEMBERS*), Philadelphia, Pa.

IN recent years the problem of simultaneous cooling and dehumidification of air with cold surfaces has received the attention of a considerable number of experimenters. The results of these studies have been to evolve certain concepts and fundamental expressions that relate in an instantaneous manner the variables in this particular heat flow mechanism. Of these, the concept of surface temperature, Lewis'[1] straight line law, Merkel's[2] derivation of the proper heat flow potential, and Goodman's[3] derivation of the condition curve are outstanding and are widely accepted as basic in the industry.

It is a major purpose of the authors to develop from this background a rational and convenient method of anticipating the area of fin and tube surface required to remove from air a specific total heat quantity in which both sensible and latent heat are transferred. Heretofore, one of the difficulties in the application of fundamental expressions to accomplish this has resulted from the complex equations necessary to relate total heat and temperature at saturation over a wide range of temperature. The method to be presented, by employing a straight line relation between these two quantities over a narrow range, arrives at a final expression in which the required area is determined directly as a function of the mean difference between the total heat of the air and total heat corresponding to saturation at surface temperature. Of further interest is the presentation of a surface temperature chart that permits a graphical solution for the condition at which dehumidification begins for that type of problem in which a portion of the surface is in a dry condition.

Throughout the paper consideration is given only to the counterflow type of problem employing water as the cooling medium. It will be evident later that the method derived is also applicable to cases involving a constant temperature refrigerant and to parallel flow.

In order to employ the essential concept of surface temperature, it is necessary to idealize fin and tube surface as shown in Fig. 1 to consist of a resistance to heat flow which is constant for a given surface. This permits the assumption that the temperature of the surface is constant at any point in a plane through the coil at right angles to the air flow. For a given water velocity the resistance to heat flow of the water film is also constant and the sum of the fin, tube and water film resistances may be expressed as a constant R.

Between the air and the surface is a film of stagnant air offering resistance to heat flow. This resistance will be expressed in reciprocal form as $1/f_g$ where f_g is the airside coefficient of sensible heat transfer. The value of f_g is assumed constant for a given air velocity.

The mathematical and experimental works previously mentioned have indicated that the droplets of water condensed from the air on the surface have little, if any, effect on either f_g or R. In this paper it is assumed that surface moisture does not affect f_g or R.

The transfer of heat from the warm moist air to the cold surface and thence through the fins and tubes to the cold water may best be analyzed by dividing the process into two distinct steps, (1) from the air to the wetted surface and (2) from the surface through the fins and tubes to the water.

The total heat lost by the air in passing over an element of wetted area dAw is given by

$$dH_T = \frac{f_g}{s}(h - h_S)\,dAw \qquad \ldots\ldots(1)$$

This equation which was originally derived by Merkel,[4] combines sensible heat transfer due to temperature difference and latent heat transfer due to vapor pressure difference into a single equation which states that the rate of simultaneous sensible and latent heat flow depends upon the difference between the total heat of the air flowing over the surface and the total heat corresponding to saturation at the surface temperature.

The total heat transferred from the element of area dAw through the fins and tubes to the water is given by

$$dH_T = \frac{1}{R}(t_S - t_R)\,dAw \qquad \ldots\ldots(2)$$

Where R is as defined above the composite resistance to heat flow imposed by the fins and tubes and the internal water film.

Equating (1) and (2) and rearranging

$$\frac{t_S - t_R}{h - h_S} = \frac{f_g R}{s} = C \qquad \ldots\ldots(3)$$

This is essentially the same form as originally derived by Goodman.[5]

*Research Engr., John J. Nesbitt, Inc.
**Research Engr., John J. Nesbitt, Inc.
[1]*Industrial and Engineering Chemistry*, C. S. Keevil and W. K. Lewis, Vol. 20, No. 10, October, 1928, p. 1058.
[2]Merkel, Zeit, Ges. Kalte Industrie, Vol. 34, July, 1927, p. 117.
[3]Dehumidification of Air with Coils by William Goodman (*Refrigerating Engineering*, Vol. 32, No. 4, October, 1936, p. 225).
Presented at the 46th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Cleveland, O., January, 1940.

[4]Loc. Cit. Note 2.
[5]Loc. Cit. Note 3.

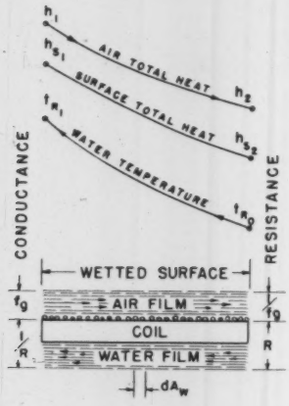

Fig. 1—Idealization of fin and tube type surface

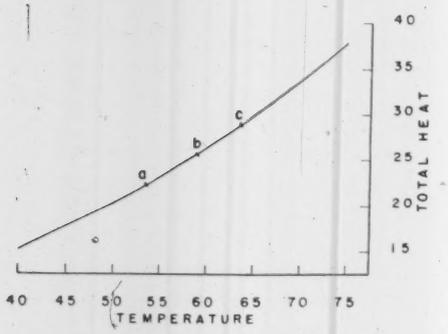

Fig. 3—Total heat vs. temperature

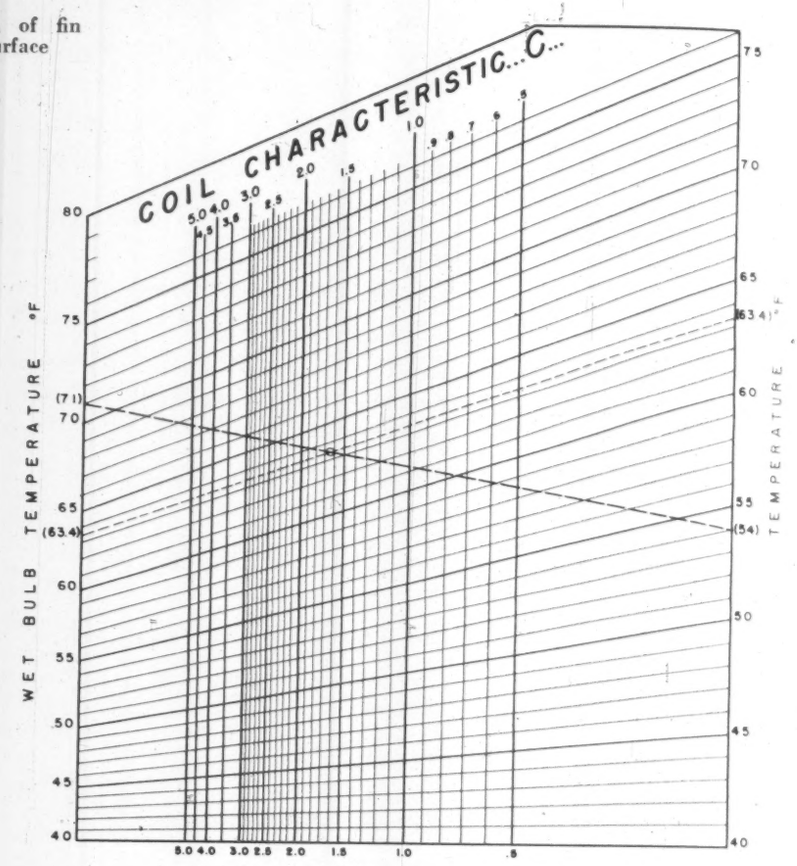

Fig. 2—Surface temperature chart

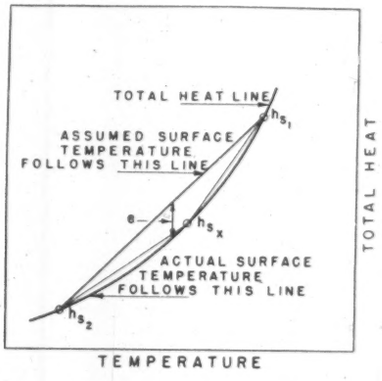

Fig. 4—Exaggerated scale of total heat vs. temperature

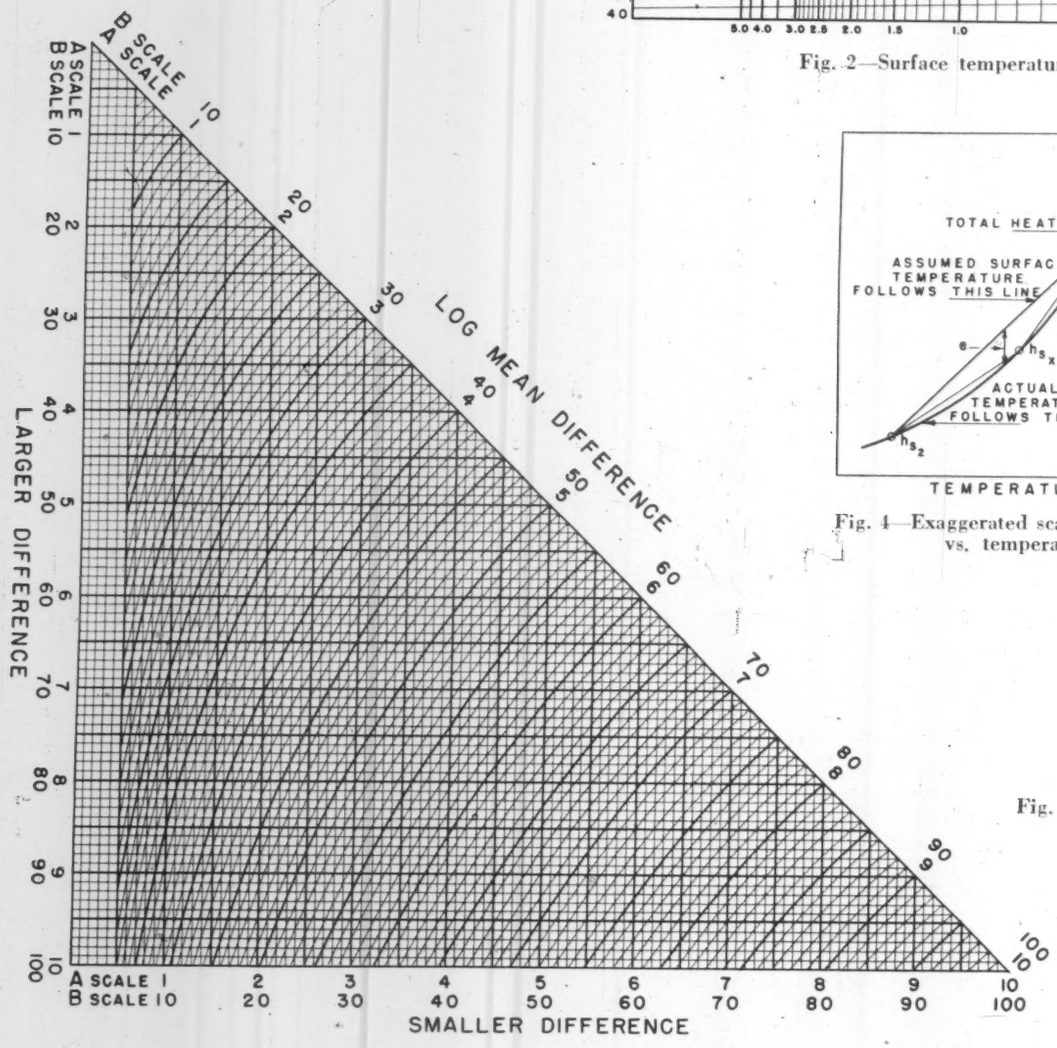

Fig. 5—Chart of logarithmic mean difference

The value of s, the humid specific heat of air, may be taken as constant for the range of comfort cooling. For a given coil, air velocity and refrigerant velocity, f_g and R are constant and thus C is a constant coil characteristic for given conditions. It should be noted that Equation (3) refers to the wetted portion of the surface. Through the wetted portion, moisture is condensing on the surface from the air. Therefore, h_s is the total heat at saturation at the surface temperature t_s.

The significance of Equation (3) is that for a particular coil employing specific air and water quantities, the value of C will be fixed and hence the interdependent values of the total heat of the air h, the surface temperature t_s and the total heat h_s corresponding to saturation at the surface temperature may be evaluated for any point in the passage of air through the coil. Note that t_R, the temperature of the water at any point, is essentially a function of h, being related to it through the ratio of the air and water quantities.

Direct calculation of t_s and h_s for specific values of h and t_R is not feasible and resort must be made to either a trial and error solution employing psychrometric tables or to a chart depicting the proper relationship between these values.

Fig. 2 illustrates a surface temperature chart that permits a graphical solution of Equation (3). The scale at the left is plotted to even increments of total heat to which the corresponding wet-bulb temperatures have been assigned. It may be regarded as the scale expressing h and h_s. The right hand scale is plotted to even increments of temperature and represents t_s and t_R. The differences $(h\text{-}h_s)$ and $(t_s\text{-}t_R)$ are then represented by intervals on the respective total heat and temperature scales.

As proved in Appendix 1, a line connecting the wet-bulb temperature of the air on the left hand scale and the water or refrigerant temperature corresponding on the right hand scale will intersect the C line for the coil at the prevailing surface temperature as read from the slanting lines. This can best be illustrated by a specific example.

Example 1: For a given surface $f_g = 10$ $R = 0.04$
Find (*a*) the value of C
(b) the surface temperature t_s when the refrigerant temperature $t_R = 54$ F and the wet-bulb temperature $t' = 71$ F

Solution: (*a*) Using Equation (3) Let $s = 0.245$

$$C = \frac{10 \times 0.04}{0.245} = 1.63$$

(*b*) Using the surface temperature chart in Fig. 2 spot the refrigerant temperature $t_R = 54$ F on the right hand scale and the air wet-bulb temperature $t' = 71$ F on the left hand scale. Draw the line intersecting these two points. Where this line intersects the vertical line $C = 1.63$, (interpolate between 1.6 and 1.7) follow the slanting directrix lines to either the right hand or left hand scale and read the surface temperature by interpolation as 63.4 F.

In *Example 1*, the assumption is made that the dew-point temperature of the air at the condition of 71 F wet-bulb is equal to or above the surface temperature 63.4 F; otherwise moisture will not be condensed on the surface and neither Equation (3) nor Fig. 2 will truly represent the actual surface temperature. That a coil may readily operate to remove only sensible heat from the air for a portion at the entering air side and both sensible and latent heat for the remainder is a matter of common experience. Dehumidification will take place in a coil only where the dew-point temperature is equal to or above the surface temperature.

For the present, discussion will be confined to means of calculating wetted surface areas only, whether this area represents a complete coil or only a portion thereof. Later, consideration will be given the treatment of the problem involving dry surface in combination with wetted surface.

Determination of the Wetted Surface Area Required

Equation (1), as repeated, states that the potential for the combined sensible and latent heat flow from the air to the wetted surface is the difference between the total heat of the air flowing over the surface and the total heat corresponding to saturation at the surface temperature.

$$dH_T = \frac{f_g}{s}(h - h_s)\, dAw$$

Since both h and h_s are variable this expression in the form shown is suitable only for the calculation of the differential area required to remove a differential quantity of heat with the instant potential $(h\text{-}h_s)$. To integrate Equation (1) and therefore to sum up the area required to remove from the air a specific total heat quantity, it is first necessary to relate h_s to h.

Restating Equation (3)

$$\frac{t_s - t_R}{h - h_s} = C$$

Rearranging Equation (3) and differentiating

$$C\,dh + dt_R = C\,dh_s + dt_s \qquad (4)$$

The total heat given up by the air in flowing over the surface is at all points equal to the heat gained by the water for the interval considered.

$$W_s\,dh = -W_R\,dt_R \qquad (5)$$

Substituting Equation (5) in Equation (4) and rearranging

$$\left(C - \frac{W_s}{W_R}\right) dh = C\,dh_s + dt_s \qquad (6)$$

Equation (6) shows that any relation between h and h_s has its origin in a relation between h_s and t_s, that is, a relation is first required that expresses total heat in terms of temperature at saturation for the range over which the surface temperature varies.

It may be observed either from psychrometric tables or from the total heat temperature line itself that the exact relation between h_s and t_s over a wide range is quite complex. The substitution of such a relationship in Equation (6) so complicates Equation 1 that complete integration is extremely difficult if not impossible.

To arrive at an integrable form of Equation (1) it is, therefore, necessary to employ a relationship between h_s and t_s that may be accepted as true over a limited range. In Fig. 3, total heat is plotted against temperature at saturation for the range of surface temperatures normally encountered. For limited ranges on this line

such as a to b or b to c, a linear relationship between h_s and t_s closely approximates the true values. The degree of accuracy with which the total heat values as given by the chord between the end points checks the actual values is affected by the range employed. For the present, no attempt will be made to define the extent of the acceptable range but attention will be directed toward solution of Equation (1) employing this linear relation.

$$h_s = m + nt_s \quad \text{for a limited range of } t_s \ldots\ldots (7)$$

$$\text{Differentiating} \quad dh_s = n dt_s \ldots\ldots (8)$$

Substituting (8) in (6)

$$dh_s = \frac{C - \frac{W_g}{W_R}}{\frac{1}{n} + C} dh = E dh \ldots\ldots (9)$$

This provides the relationship between h and h_s necessary for integration of Equation (1).

As shown in Appendix II Equation (1) may now be integrated to arrive at the final expression for the wetted surface area.

$$A_W = \frac{H_T s}{f_g MHD} \ldots\ldots (10)$$

Where A_W = wetted surface area required

H_T = total heat in Btu per hour removed from the air in its passage through the coil

$= W_g (h_1 - h_2)$

h_1 is initial total heat of the air

h_2 is final total heat of the air

MHD = Log mean heat difference between the air and surface.

$$= \frac{\theta_L - \theta_S}{2.3 \log_{10} \frac{\theta_L}{\theta_S}}$$

θ_L = Larger total heat difference between the air and surface

θ_S = Smaller total heat difference between the air and surface.

Example 2: Find the area required to cool air from an entering wet-bulb temperature of 70 F to a final wet-bulb temperature of 55 F using chilled water entering at 45 F and leaving at 55 F. The air velocity is 500 fpm standard air and the airside coefficient f_g is 9.7. The water velocity is 2 fps and the resistance of the water film and copper R is 0.0378.

Using Equation (3) $\quad C = \frac{9.7 \times 0.0378}{0.245} = 1.5$

Referring to the surface temperature chart as shown in Fig. 2

t_{S1} = 63.0 F corresponding to t'_1 = 70 F and t_{R2} = 55 F

t_{S2} = 49.6 F corresponding to t'_2 = 55 F and t_{R1} = 45 F

h_1 = 33.51 $\quad h_{S1}$ = 28.22 $\quad \theta_1$ = 5.29

h_2 = 23.04 $\quad h_{S2}$ = 19.97 $\quad \theta_2$ = 3.07

$$MHD = \frac{5.29 - 3.07}{2.3 \log_{10} \frac{5.29}{3.07}} = 4.08$$

From Equation (10)

$$A_W = \frac{H_T s}{f_g MHD}$$

$$H_T = W_g (h_1 - h_2)$$

$$= \frac{500 \times 60}{13.35} (33.51 - 23.04)$$

$$= 23{,}600 \text{ Btu per hour}$$

$$\text{Then } A_W = \frac{23{,}600 \times 0.245}{9.7 \times 4.08} = 145.5 \text{ sq ft wetted surface per sq ft of face area}$$

In the application of the foregoing to specific wetted surface problems there naturally arises the question of the magnitude of the error involved in the calculation of surface areas based on the assumed straight line relationship between h_s and t_s for a particular surface temperature range. This can best be answered by first referring to Fig. 4, that represents the total heat temperature line and the assumed chord to an exaggerated scale.

With air entering a coil at h_1 and leaving at h_2, the corresponding wetted surface total heats will be h_{s_1} and h_{s_2} respectively and the chord connecting h_{s_1} and h_{s_2} will represent the straight line assumption between these two end points. It is evident that the total heat as anticipated by the line will (except at the end points) exceed the true total heat for a particular surface temperature by a variable quantity e. The maximum value of e depends on the range over which the assumption is made or approximately on the difference $(h_{s_1}\text{-}h_{s_2})$. Since the chord lies above the true h_s line, the true $(h\text{-}h_s)$ in Equation (1) is reduced by e. The comparative error introduced in the area required is not only a function of e but depends also upon the potential $(h\text{-}h_s)$. Thus, if $(h\text{-}h_s)$ is small then the deviation between the true and the assumed value of h_s will bear an important relation to the area while if $(h\text{-}h_s)$ is large the significance of the deviation is reduced.

It may be noted that if an intermediate true surface temperature h_{sx} corresponding to an air condition h_x between h_1 and h_2 is established, then calculation of the individual areas for the steps h_1 to h_x and h_x to h_2 will greatly improve the accuracy. The final criterion for any problem is to divide the surface temperature (and air condition) range into a number of individual steps such that the total area representing the sum of the individual areas differs by a negligible quantity from the total area obtained as a result of one less step.

The number of individual steps necessary to provide a degree of accuracy acceptable in engineering calculations is, as implied previously, dependent both upon the range of surface temperatures subtended by the initial and final conditions and the mean heat difference for these conditions.

For surface temperatures between 40 F and 75 F a single solution of Equation (10) will usually check within 3 per cent of a point by point solution where the surface temperature range does not exceed 12 F and the mean heat difference corresponding to the initial and final conditions is greater than 3. Since the chord lies above the true saturation line the error obtained is always of a positive nature. A two step solution will provide approximately the same degree of accuracy for the same mean heat difference over twice the surface temprature range. These boundary conditions are necessarily approximate for it will be apparent that anticipation of the error involved in any problem essentially rests with a multi-point solution.

However, in the range of cooling and dehumidifying problems normally encountered in comfort air conditioning practice a single step solution for the narrower ranges and a two step solution for the wider ranges will prove sufficiently accurate. It only becomes necessary to employ more than a two step solution where the quantity of surface involved is very large and the mean heat difference is of the order of 1½ or less.

Example 2 shows the procedure employed in a single step solution.

The simplest approach to a two step solution is to divide the total heat to be removed in the wetted surface portion of a coil into two equal parts and then to calculate the total area from the respective mean heat differences so established. Thus, if the total heat of the entering air is h_1 and the total heat of the leaving air is h_2, an intermediate total heat h_m is selected such that

$$(h_1 - h_m) = (h_m - h_2)$$

The corresponding surface total heats as determined from the surface temperature chart (Fig. 2) will be h_{s1}, h_{sm} and h_{s2}.

Then combining the expressions for the individual areas as given by Equation (10)

$$A_w = A_{w1} + A_{w2} = \frac{H_T s}{2 f_g}\left(\frac{1}{MHD_1} + \frac{1}{MHD_2}\right) \quad \ldots\ldots (11)$$

Note that:

$$\frac{H_T}{2} = W_g (h_1 - h_m) = W_g (h_m - h_2)$$

MHD_1 is determined by the conditions h_1, h_m, h_{s1} and h_{sm} while MHD_2 is determined by h_m, h_2, h_{sm} and h_{s2}.

Fig. 5 expressing logarithmic mean heat difference in terms of the larger and smaller differences simplifies the determination of this value.

Example 3: Find the area required to cool air from an entering wet-bulb temperature of 78 F to a final wet-bulb temperature of 55 F using chilled water entering at 45 F and leaving at 60 F.

The air velocity is 300 fpm of standard air and the corresponding air side coefficient $f_g = 6.48$.

The water velocity is 2 fps and the resistance of the water film and copper, R, is 0.0378.

Using Equation (3) $\quad C = \dfrac{6.48 \times 0.0378}{0.245} = 1$

For a two step solution a point midway between h_1 and h_2 is assumed. Since $h_1 = 40.64$, $h_2 = 23.04$, $h_m = 31.84$ and t'_m corresponding is 67.9.

The corresponding water temperature $t_{Rm} = \dfrac{45 + 60}{2} = 52.5$ F

From the surface temperature chart,

t_{s1}	= 68.4 F	corresponding to t'_1	= 78 F	and t_{R2}	= 60 F
t_{sm}	= 58.9 F	corresponding to t'_m	= 67.9 F	and t_{Rm}	= 52.5 F
t_{s2}	= 48.6 F	corresponding to t'_2	= 55 F	and t_{R1}	= 45 F
h_1	= 40.64	h_{s1}	= 32.24	θ_1	= 8.40
h_m	= 31.84	h_{sm}	= 25.46	θ_m	= 6.38
h_2	= 23.04	h_{s2}	= 19.44	θ_2	= 3.60

Referring to Fig 5. MHD_1 between 8.40 and 6.38 is equal to 7.35 and the MHD_2 between 6.38 and 3.60 is equal to 4.86.

H_T per sq ft of face area is equal to

$$300 \times \frac{60}{13.35} \times (40.64 - 23.04) = 23{,}800 \text{ Btu per hour}$$

From Equation (11)

$$A_w = \frac{23{,}800 \times 0.245}{2 \times 6.48}\left(\frac{1}{7.35} + \frac{1}{4.86}\right) = 154 \text{ sq ft of wetted surface per square foot face area.}$$

Note: If the above example were solved using but one step, the MHD between the end points $\theta_1 = 8.40$ and $\theta_2 = 3.50$ would be 5.66. Substituting this value in Equation (1)

$$A_w = \frac{23{,}800 \times 0.245}{6.48 \times 5.66} = 159 \text{ sq ft per square foot of face area.}$$

Or an error of + 3 per cent would have been made if one step were used instead of two steps.

Graphical Determination of the Air and Refrigerant Conditions at the Point Where Dehumidification Begins

If the surface temperature at the point where the air enters a coil is higher than the entering air dew-point temperature, then for a portion of the coil only sensible heat is transferred. The equations for wetted surface derived in this paper apply only to simultaneous sensible and latent heat transfer. The well known equations for sensible heat transfer do not apply for combined cooling and dehumidification. As might be expected, large errors must result if the presence of considerable dry surface is neglected and the coil area computed as if all wetted or all dry. The relation between the entering air dew-point temperature and the entering surface temperature should, therefore, be observed early in the solution of a cooling coil problem. If dry surface exists, the coil must be treated as if there were two coils in series; one consisting of all dry surface and the other of all wetted surface. Such a condition is illustrated in Fig. 6.

In order to calculate separately the dry and wetted surface, it is necessary to determine the air and refrigerant conditions at the point where dehumidification begins. The surface temperature at this point is known, that is, it must be just equal to the entering air dew-point temperature in order for dehumidification to just begin. The corresponding air and refrigerant condition may be determined graphically from the surface temperature chart and the psychrometric chart.

The heat lost by the air in flowing over any portion of a coil is equal to the heat gained by the water. Thus referring to the wetted portion of the coil in Fig. 6,

$$W_g (h_1 - h_2) = W_R (tr_1 - tr_0) \quad \ldots\ldots (12)$$

$$\text{and } \frac{tr_1 - tr_0}{h_1 - h_2} = \frac{W_g}{W_R} = B \quad \ldots\ldots (13)$$

For any given air and water quantities, B is a constant.

Equation (3) may be partially restated as follows:

$$\frac{f_g R}{s} = C \quad \ldots\ldots (14)$$

For a given coil, air velocity and water velocity, C is a constant.

Fig. 7 represents a skeleton surface temperature chart. The only slanting directrix line shown is that of the surface temperature which is equal to the entering air dew-point temperature. The point P_1 on this directrix line is the intersection of the C line from Equation (14). From the point P_1 a line is drawn to t'_2 the leaving air wet-bulb temperature. The value of B from Equation 13 is added to C and the point P_2 determined where the vertical line $B + C$ intersects the line (P_1, t'_2). The entering refrigerant temperature t_{Ro} is spotted on the right hand scale and a line drawn through P_2 to the left hand scale. This point is t'_1, the air wet-bulb temperature at the point where dehumidification begins. From t'_1, a line is drawn through P_1 to the right hand scale. This point is t_{R1}, the refrigerant temperature at the point

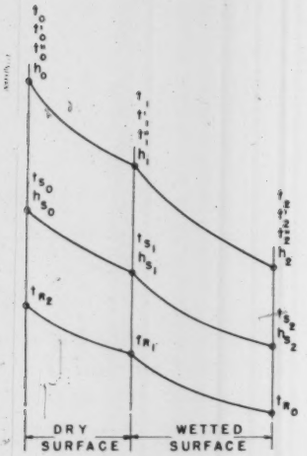

Fig. 7 (*above*) — Skeleton surface temperature chart

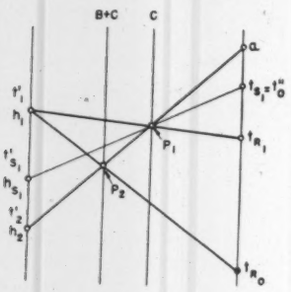

Fig. 6 (*left*)—Diagram of coil surface showing dry and wetted surfaces

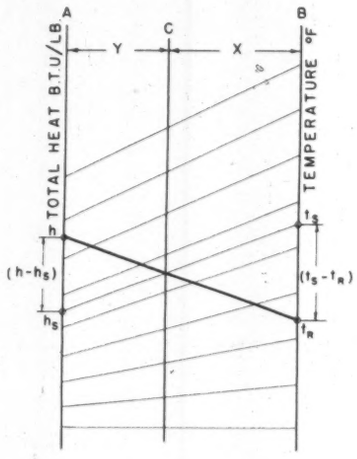

Fig. 8 (*left*)—Basic surface temperature chart

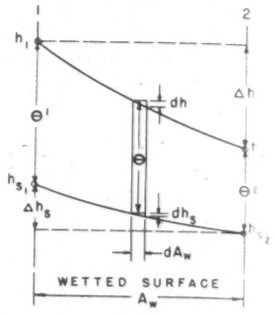

Fig. 9 (*above*)—Diagram of heat flow for wetted surface

where dehumidification begins. When the air dew-point and wet-bulb temperatures are known, the corresponding dry-bulb temperature is read from a psychrometric chart.

The proof of this graphical method is as follows:

From Fig 7, $\dfrac{a - t_{R0}}{h_1 - h_2} = B + C$ ..... (15)

also $\dfrac{a - t_{R1}}{h_1 - h_2} = C$ ..... (16)

Equating (15) and (16)

$$\frac{t_{R1} - t_{R0}}{h_1 - h_2} = B$$

This is Equation (13).

Determination of Dry Surface Area

The amount of dry surface required is determined by the familiar equation

$$A_D = \frac{H_D}{U\,MTD} \quad \text{..... (17)}$$

Where H_D is the sensible heat lost by the air in flowing over the dry surface and may be calculated from the equation

$$H_D = W_g\, s\, (t_0 - t_1) \quad \text{..... (18)}$$

and U is the over-all heat transfer coefficient in Btu per square foot air side area per hour per degree logarithmic mean temperature difference between the air and refrigerant. The value of U is found from the equation

$$U = \frac{1}{R + \dfrac{1}{f_g}} \quad \text{..... (19)}$$

The mean temperature difference is found by the equation

$$MTD = \frac{D_L - D_S}{2.3\, Log_{10} \dfrac{D_L}{D_S}} \quad \text{..... (20)}$$

Where D_L is $(t_0 - t_{R2})$ or $(t_1 - t_{R1})$ whichever is larger and D_S is $(t_1 - t_{R1})$ or $(t_0 - t_{R2})$ whichever is smaller.

Conclusions

A surface temperature chart is presented which permits the direct determination of surface temperature from the air wet-bulb temperature, the refrigerant temperature and the coil characteristic. A method is presented for the graphical determination on the surface temperature chart of the air and refrigerant conditions at the point where dehumidification begins.

The logarithmic mean total heat difference is derived as the potential causing heat flow from warm moist air to cold wetted surface. The relation between total heat at saturation and temperature is investigated. Equations are presented for computing the amount of surface required to cool and dehumidify air to specified conditions of total heat.

Appendix I

The Surface Temperature Chart

Equation (3) is

$$\frac{t_S - t_R}{h - h_S} = C$$

In Fig. 8, two parallel lines A and B are divided into increments respectively a to the inch and b to the inch. The A scale is marked as total heat and the B scale is marked as dry-bulb temperature. From the B scale, the slanting directrix lines are drawn to the corresponding total heats at saturation on the A scale.

If values of h, h_s, t_s and t_R are spotted on the chart as shown and connected by the lines $(h,\ t_R)$ and $(t_s,\ h_s)$, two similar triangles will be formed intersecting at the line C, and from the geometry of the chart

$$\frac{a\,(t_S - t_R)}{b\,(h - h_S)} = \frac{X}{Y} = C$$

Thus lines representing C may be drawn on the chart. In the final form of the chart the total heat values are omitted and only the wet-bulb temperatures are shown on the left hand scale.

Appendix II

Derivation of $A_w = \dfrac{H_T \times S}{f_g \times MHD}$

From Equation (1) of the text:

$$dH_T = W_g dh = \frac{f_g}{s}\,(h - h_S)\,dA_w \quad \text{..... (1)}$$

From Equation (9) of the text:

$$dh_S = E\,dh \quad \text{..... (2)}$$

Referring to Fig 9:

Let $\theta = h - h_S$ ..... (3)

then $d\theta = dh - dh_S$ ..... (4)

Substituting Equation (2) in Equation (4) and rearrang-

ing $dh = \frac{d\theta}{1-E}$ ..... (5)

then from Equations (1) and (5)

$$\frac{d\theta}{\theta} = \frac{f_g\,(1-E)}{s\quad W_g}\,dA_w \quad \text{..... (6)}$$

Integrating between limits of θ_1 and θ_2

$$Log_e \frac{\theta_1}{\theta_2} = \frac{f_g\,(1-E)}{s\quad W_g} A_w \quad \text{..... (7)}$$

Multiply both sides of Equation (7) by $\frac{\theta_1 - \theta_2}{Log_e \frac{\theta_1}{\theta_2}}$

$$\theta_1 - \theta_2 = \frac{f_g\,(1-E)}{s\ W_g} A_w \frac{\theta_1 - \theta_2}{Log_e \frac{\theta_1}{\theta_2}} \quad \text{..... (8)}$$

Referring to Fig. (9)

$\theta_1 - \theta_2 = \triangle h - \triangle h_s$ ..... (9)

and from Equation (2) $\triangle h_s = E \triangle h$ ..... (10)

Then from Equations (9) and (10)

$$\triangle h = \frac{\theta_1 - \theta_2}{1-E} \quad \text{..... (11)}$$

Since $H_T = W_g \triangle h$ ..... (12)

$$H_T = W_g \frac{\theta_1 - \theta_2}{1-E} \quad \text{..... (13)}$$

Substituting Equation (8) in Equation (13)

$$H_T = \frac{f_g}{s} A_w \frac{\theta_1 - \theta_2}{Log_e \frac{\theta_1}{\theta_2}} \quad \text{..... (14)}$$

By definition $MHD = \frac{\theta_1 - \theta_2}{Log_e \frac{\theta_1}{\theta_2}}$ ..... (15)

Then

$$A_w = \frac{H_T \times S}{f_g \times MHD} \quad \text{..... (16)}$$

It should be noted that three cases for this derivation exist—

Case 1 where $\theta_1 > \theta_2$ as shown

Case 2 where $\theta_1 = \theta_2$ $MHD = \theta_1 = \theta_2$

Case 3 where $\theta_1 < \theta_2$ The respective positions of θ_1 and θ_2 are reversed.

To cover Cases 1 and 3

$$MHD = \frac{\theta_L - \theta_S}{Log_e \frac{\theta_L}{\theta_S}}$$

where

θ_L = Larger heat difference.

θ_S = Smaller heat difference.

Symbols

A = Total square feet airside area, i.e., the total external surface of the fins and tubes exposed to air flow.

A_D = Square feet dry airside surface.

A_w = Square feet wetted airside surface on which moisture is condensing.

f_g = Sensible heat transfer coefficient through air film in Btu per hour per degree Fahrenheit per square foot airside surface.

W_g = Total weight of air, pounds per hour.

W_R = Total weight of refrigerant, pounds per hour.

H_D = Sensible heat lost by total weight of air flowing over dry surface, Btu per hour.

H_T = Total heat lost by total weight of air flowing over wetted surface, Btu per hour.

h = Total heat content of air vapor mixture per pound of dry air. Btu values are from Goodenough's tables and include the sensible heat of the air plus the heat of vaporization and the heat of superheat of the water vapor. Wet-bulb temperature is taken as the sole index of total heat.

s = Humid specific heat of air vapor mixture, Btu per pound of dry air per degree Fahrenheit.

t = Dry-bulb temperature, degrees Fahrenheit.

t' = Wet-bulb temperature, degrees Fahrenheit.

t'' = Dew-point temperature, degrees Fahrenheit.

R = Resistance to heat flow through fins, tubes and refrigerant film per square foot airside.

U = Over-all sensible heat transfer coefficient. Btu per hour per square foot airside per degree between air and refrigerant.

Subscripts

$_0$ = Refers to the initial condition of air or refrigerant and the condition of the surface where the air enters.

$_1$ = Refers to the condition of air, surface and refrigerant at the point where dehumidification begins.

$_2$ = Refers to the final condition of air or refrigerant and the condition of the surface where the air leaves.

$_R$ = Refers to the refrigerant.

$_S$ = Refers to the surface.

$_g$ = Refers to the air.

Miles of Housing — or Fortifications

More than a quarter of a billion dollars are being put into circulation through building trades materials industries by the present slum-clearance and low-rent housing program of the United States Housing Authority. These materials, which are itemized in an analysis issued by the USHA are the products of more than a score of American industries.

Thousands of tons of steel, iron, other metals and alloys, bricks, cement and stone, and millions of feet of lumber are included in these estimates, together with huge purchases of concrete products, electrical materials, fixtures and machinery, heating and ventilating equipment and materials, flooring, roofing, and movable dwelling equipment. The estimated total cost of construction materials for the entire program of 160,000 dwelling units is $280,000,000. This total cost approximates the respective expenditures of France and Germany in constructing their famed Maginot and Siegfried Lines and the materials are much the same in kinds and quantities.

The Maginot Line in France is about 125 miles long and its estimated cost was approximately $2,000,000 per mile. No data are available concerning the Siegfried Line in Germany, but the cost of construction is estimated at somewhat more than that of the Maginot Line. Materials used to construct these miles of steel and concrete forts and trenches, and much of their equipment other than armament, were the same as are being used in the USHA program to build safe and sanitary homes for low-income American families.

Commenting on the economic benefits of the large use of the products of American industries, it has been stated that millions of dollars have been injected into the bloodstream of American business flow through virtually every artery of commercial and professional, as well as industrial enterprise in our country.

The wages paid to the workers in the more than a score of industries, from which these construction materials are being purchased, for example, stimulate local business in the hundreds of communities in which the mills and other plants are located. These millions of dollars going into pay envelopes are translated into more sales for local shops, garages and theaters and fees for doctors, dentists and other professional practitioners.

The present USHA slum-clearance and low-rent housing program provides for construction in 168 communities of 160,000 safe and sanitary homes to rehouse that number of low-income families, or about 640,000 persons, now living in sub-standard structures and squalid environment.

More than 40 per cent of this construction now is under way and about 21 per cent will be completed by June 1940.

Pittsburgh Experiment Station of the U. S. Bureau of Mines where the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS is located

Analysis of the Factors Affecting Duct Friction

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THE results of a study of frictional resistance to the flow of air in ducts were presented at the 45th Annual Meeting of the Society in January, 1939. This paper[1] contained air flow-friction data applying to straight round ducts with and without joints as found under experimental conditions. Since the publication of these findings the Laboratory has been interested in a mathematical analysis of the friction flow relationships, with a view of developing mathematical formulae which may be used in satisfactorily expanding the test data from a few sizes of ducts to a wide range of duct sizes and flow conditions. This analysis, based upon a modified form of the Fanning equation and the friction factor —Reynolds Number relationships for ducts with no joints and ducts with 40 joints per 100 ft length of run, is presented herein.

The loss of pressure due to the frictional resistance to flow of fluids in pipes and ducts is often expressed by the Fanning formula as:

$$H = \frac{fLV^2}{2gD} \qquad \ldots\ldots (1)$$

where H = pressure loss in feet of the fluid
f = friction factor
L = length of duct in feet
V = velocity in feet per second
g = acceleration due to gravity = 32.2 ft per sec^2
D = diameter of duct in feet

In this formula the friction factor, f is expressed in terms of the Reynolds Number and for ducts or pipe of given characteristics as regards to material and smoothness of construction this factor has generally been assumed to be constant, regardless of size. An average value of 0.0149 taken from the test results for the friction factor in the earlier Laboratory report[2] led to the development of the following formula:

$$H = \frac{0.765\,fL\,(V/4000)^{1.84}}{D^{1.20}} \qquad \ldots\ldots (2)$$

where the different symbols, except for H which is given in inches of water and for V which is expressed in feet per minute, have the same significance as indicated for the Fanning formula previously stated. For ducts with 40 joints per 100 ft length of run the average value of the friction factor as taken from the test data was 0.0192, giving the following:

$$H = \frac{0.75fL\,(V/4000)^{1.84}}{D^{1.31}} \qquad \ldots\ldots (3)$$

Equations (2) and (3), resulting from a ready analysis of the Laboratory's data as presented in the earlier report, offer an opportunity for an extended analysis as regards the numerical value of the velocity exponent

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[1]ASHVE RESEARCH REPORT No. 1105—Frictional Resistance to the Flow of Air in Straight Ducts, by F. C. Houghten, J. B. Schmieler, J. A. Zalovcik and N. Ivanovic. (ASHVE TRANSACTIONS, Vol. 45, 1939.)

Presented at the 46th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Cleveland, Ohio, January, 1940.

[2]Loc. Cit. Note 1.

and to a greater extent as regards the evaluation of the friction factor f.

The velocity exponent 1.84 was taken as the numerical average of the observed values for a large number of tests. These values have since been subjected to a statistical treatment, wherein the results of a number of tests on round, shop-made ducts without joints were plotted on log-log paper, and the velocity exponents in the equations showing variation of pressure drop with velocity were measured and found to be as follows:

4-IN. ROUND	8-IN. ROUND	24-IN. ROUND
1.75	1.92	1.64
1.85	1.90	1.83
1.81	1.88	1.89
1.68	1.81	1.80
.....	1.87	1.81
.....	1.91	.....
.....	1.86	.....
.....	1.87	.....
.....	1.79	.....
.....	1.80	.....
.....	1.80	.....
.....	1.87	.....

These exponents were found to have the average value of 1.826 and when treated statistically the standard deviation σ was found to be 0.0698, indicating that the most probable value for the average velocity exponent is 1.826 ± 0.027. This means that 9 groups out of 10, each containing approximately the same number of observations as this group, will yield an average value of the velocity exponent within the limits given. Although some authorities have used an exponent of 2.0, it can be seen that the test results never gave a value above 1.92. Most recent investigators show a fractional velocity exponent for ducts of roughness comparable to sheet metal. Noteworthy among these is the work of Kemler[3] in which are found data by numerous experimenters throughout the country on resistance to flow of various fluids through many types of pipes and ducts. In nearly all cases, the velocity exponent was lower than 2.0. Kemler's work shows velocity exponents by various investigators for galvanized ducts of 1.67, 1.77, 1.82, 1.83, 1.85, and 1.89. The 1938 edition of *Fan Engineering*[4] gives 1.82 for smooth, straight galvanized ducts and clean steel and wrought iron pipes. THE ASHVE GUIDE for the past number of years has shown a value of 1.86 for the exponent. These disagreements may be due to variations in the test set-up, physical differences in the material used, method of analysis, nature of turbulent flow, and other factors.

Fig. 1 shows the relation between the friction factor and the Reynolds Number for the three sizes of round ducts studied at the Laboratory and presented in the earlier paper.[4] As pointed out therein it is seen that rather than being independent of duct size definite curves result for each duct size. The reason for this lies in the fact that the roughness of the sheet metal is more or less constant rather than proportional to the duct size. Also, it will be seen that the shape of the curves makes it impossible to use a single value of f in equation (1) for calculating pressure drops, even for a single size of duct of any particular design, but that this value must be some function of the Reynolds Number. This suggests the possibility of expressing f as some function of the Reynolds Number and duct size, or as some function of the two expressions.

The friction factor—Reynolds Number relationship is commonly expressed as

$$f = \frac{K}{Re^n} \text{ or } f = C + \frac{K}{Re^n} \quad \ldots\ldots (4)$$

where

f = friction factor
C = a constant
K = a constant
n = exponent (value to be determined)
Re = Reynolds Number (dimensionless)

where

$$Re = \frac{DV\rho}{\mu} = \frac{DV}{\nu}$$

D = diameter of duct in feet
V = velocity in feet per second
ρ = density of fluid in pounds per cubic foot
μ = absolute viscosity in pounds per foot second
ν = kinematic viscosity $= \frac{\mu}{\rho}$

For straight run of duct with no joints the following empirical friction formula was found to apply to the curves for all sizes in Fig. 1:

$$f = \frac{K\mu^m}{D^x V^y \rho^z} \quad \ldots\ldots (5)$$

where

K = a constant
μ = absolute viscosity in pounds per foot second
D = diameter of duct in feet
V = velocity in feet per second
ρ = density in pounds per cubic foot
m = viscosity exponent

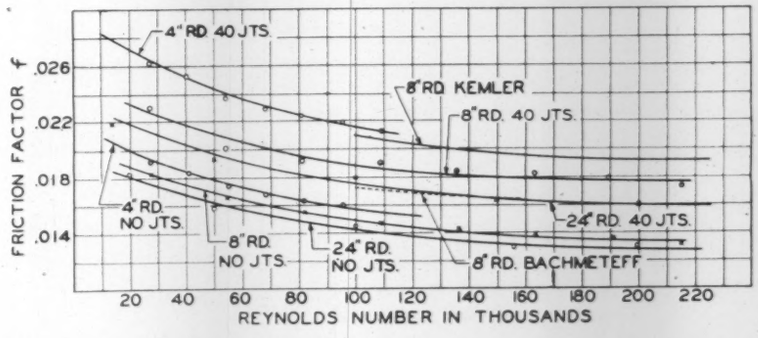


Fig. 1—Relation between the friction factor and Reynolds Number for round ducts tested without joints and with 40 joints per 100 ft length of run. Curves for data published by Kemler[5] and Bachmeteff[6] are included

[3]Study of Data on the Flow of Fluids in Pipes, by E. Kemler. (*ASME Transactions*, Hyd., Vol. 55, 1933).
[4]Loc. Cit. Note 1.
[5]Loc. Cit. Note 3.
[6]The Reynolds Number, by B. A. Bachmeteff (*Mechanical Engineering*, October, 1936).

x = diameter exponent
y = velocity exponent
z = density exponent

By maintaining all factors constant in Equation (5) and varying velocity, density, and viscosity separately and then plotting the values of f against velocity, density, and viscosity, respectively, the exponents, m, y, and z were each found to be equal to approximately 0.17.

The value of x, the diameter exponent, may be obtained through the substitution of two sets of values in the following equation, based upon Equation (5), and the substitution therein of the values found for m, y and z:

$$f = \frac{K\mu^{0.17}}{D^x V^{0.17} \rho^{0.17}} = \frac{K\nu^{0.17}}{D^x V^{0.17}} \quad \ldots\ldots (6)$$

Substituting the following values for an 8-in. duct where
$V = 800$ feet per minute
$\nu = $ (kinematic viscosity at standard conditions)
$= 16.27 \times 10^{-5}$
and $f = 0.0165$ (test data)

Equation (6) becomes

$$0.0165 = \frac{K\,(16.27 \times 10^{-5})^{0.17}}{0.667^x\,(800/60)^{0.17}} \quad \ldots\ldots (7)$$

Solving for K

$$K = \frac{0.0165\,(0.667)^x\,(800/60)^{0.17}}{(16.27 \times 10^{-5})^{0.17}} \quad \ldots\ldots (8)$$

Substituting the following values for a 24-in. duct where
$D = 2$ ft
$V = 800$ ft per minute
$\nu = 16.27 \times 10^{-5}$
and $f = 0.0135$ (test data)

Equation (6) becomes

$$0.0135 = \frac{K\,(16.27 \times 10^{-5})^{0.17}}{2^x\,(800/60)^{0.17}} \quad \ldots\ldots (9)$$

and substituting K of Equations (8) in (9) we have

$$0.0135 = \left(\frac{0.0165\,(0.667)^x\,(800/60)^{0.17}}{(16.27 \times 10^{-5})^{0.17}}\right)\left(\frac{(16.27 \times 10^{-5})^{0.17}}{2^x\,(800/60)^{0.17}}\right) \quad \ldots\ldots (10)$$

which reduces to

$$0.0135 = 0.0165\,(0.333)^x$$

or

$$x = \frac{\log 0.817}{\log 0.333} = \frac{-0.098}{-0.448} = 0.22$$

Using this value of x in solving for K in Equation (8)

$$K = \frac{0.0165\,(0.667)^{0.22}\,(800/60)^{0.17}}{(16.27 \times 10^{-5})^{0.17}} = 0.103 \quad \ldots\ldots (11)$$

and the complete friction factor equation may then be expressed as

$$f = \frac{0.103\,\mu^{0.17}}{D^{0.22} V^{0.17} \rho^{0.17}} \quad \ldots\ldots (12)$$

Table 1—Relation Between Calculated Friction Factor and Test Friction Factor for Straight Run of Duct With and Without Joints

Velocity Ft/Min	Friction Factor: Test A	Friction Factor: Test B	Friction Factor: Calculated A	Friction Factor: Calculated B	Per Cent Difference A	Per Cent Difference B
4-in. Round						
800	0.01915	0.0262	0.0193	0.0269	0.78	2.67
1600	0.01702	0.0235	0.01718	0.0240	0.94	2.13
2800	0.01564	0.0218	0.01565	0.0218	0.06	0.00
8-in. Round						
800	0.0165	0.0215	0.01654	0.0218	0.24	1.40
1600	0.0146	0.0193	0.0147	0.0194	0.80	0.52
2800	0.0134	0.0178	0.0134	0.0176	0.00	−1.12
24-in. Round						
800	0.0135	0.0165	0.0130	0.0165	−4.80	0.00
1600	0.01188	0.0144	0.01160	0.0140	−2.14	−2.87
2800	0.01041	0.0125	0.01052	0.0128	1.05	2.40

(*A*) For duct with no joints.
(*B*) For duct with 40 joints per 100 ft.

To check the accuracy of this equation the calculated results are compared with test results in Table 1. From this it can be seen that the calculated results check the test data fairly well for the three sizes of ducts at the given velocities. The maximum variation occurs for the 24-in. round duct at a velocity of 800 fpm, where the variation is 4.8 per cent. This variation is not large considering the chances for error in measurements at the low velocity. At higher velocities the calculated results check the test data very closely.

Hence, it would appear that a single equation would satisfactorily express the friction factor relationship for the three sizes of ducts listed, and for intermediate sizes, with an accuracy of at least 5 per cent.

A friction loss formula, which excludes the Reynolds Number and the friction factor entirely, may be developed by substituting the value of the friction factor, Equation (12), in the Fanning Equation (1). Changing H to inches of water and V to feet per minute, Equation (1) becomes

$$H = \frac{fL\,(V/4000)^2\,S}{D} \quad \ldots\ldots (13)$$

where

$$S = \frac{\text{density of air at test conditions}}{\text{density of air at standard conditions}}$$

Substituting Equation (12) for f in Equation (13) the friction loss formula becomes

$$H = \frac{0.103\,\mu^{0.17}\,L\,(V/4000)^2\,S}{D^{0.22} V^{0.17} \rho^{0.17} D}$$

$$= \frac{0.103\,\mu^{0.17}\,L\,V^2\,(1/4000)^{1.83}\,(1/4000)^{0.17}\,S}{D^{0.22} V^{0.17} \rho^{0.17} D} \quad \ldots (14)$$

Changing $V^{0.17}$ from feet per second to feet per minute, and substituting $S \cdot \rho$ std. for ρ and collecting terms, the equation becomes:

$$H = \frac{2.01\,(0.103)\,(0.244)\,\mu^{0.17}\,L\,(V/4000)^{1.83}\,S^{0.83}}{D^{1.22} \rho \text{ std.}^{0.17}} \quad \ldots (15)$$

For air at standard conditions and $L = 100$ ft, the equation reduces to:

$$H = \frac{1.143\ (V/4000)^{1.83}}{D^{1.22}} \qquad (16)$$

It should be emphasized that the above equations apply only to straight, round sheet metal ducts with no joints.

In Table 2 there are included a number of calculated values as well as test data values for the 4-in., 8-in. and 24-in. duct sizes which show the degree of application. Column *B* for each of the three sizes of duct gives the calculated pressure loss using the new formula, for the velocity given in Column *A*. Column *C* shows the pressure loss determined by the new formula in per cent of the average test data. It will be noted that for the three sizes of duct the average of the calculated results is 100.4 and 100.1 per cent for the first two cases and 98.5 per cent for the 24-in. duct size, with maximum variations of 1.7, 2.6, and 3.9 per cent, respectively. This is certainly a very satisfactory agreement between theoretically and experimentally determined values when the range of velocities and duct sizes are considered. Column *D* shows the pressure loss calculated by the old Laboratory formula in per cent of average test data. It will be noted that the average of the calculated values, 102.9, 100, and 94.9 per cent for the three sizes, is given. Columns *G* and *I* show variation in pressure loss for curves giving high and low values in per cent of the average test data. It should be emphasized that these variations represent extreme conditions and that few tests gave this high variation. It should also be noted that these variations may result from a wide range of factors, including errors in making physical observations, the accuracy of the instruments used, leaks in long rubber tubing connections from the static pressure taps to the manometers, air leakage from the ducts in which measurements were made, and the many possible variations in the construction of the duct itself and its erection. In the 8-in. duct study undoubtedly much of the variation is due to the many different duct set-ups used; that is, different tests in a series included duct which had been assembled and dismantled a number of times, including all the attendant misuse. In all such assembling and dismantling due care was taken to avoid misuse of the duct, but nevertheless some misuse must be assumed to have occurred. Considering all these factors, it is obvious that the agreement between the experimental and theoretical data is much better than could be hoped for in the case of practical duct installations as designed and erected throughout the country under the same set of specifications. In other words, individual workmanship, choice of materials, and erecting characteristics will probably account for greater variations than are included in the results of this study. In the last three columns, *J*, *K*, and *L*, for the 4-in. and 8-in. round ducts, are shown some comparisons between factory-made duct and the average shop-made duct values. This shows a maximum variation in pressure loss of 4.6 per cent for the 8-in. round duct and 13.8 per cent (excluding 400 fpm velocity) for the 4-in. round duct.

For straight round duct with 40 joints per 100 ft length of run the method of developing a friction loss formula was similar to that used for the duct with no joints. The friction factor equation was found empirically to be:

$$f = \frac{0.131\ \mu^{0.17}}{D^{0.30}\ V^{0.17}\ \rho^{0.17}} \qquad (17)$$

Table 1 shows the rather close agreement between test data and calculated results based on this formula. The maximum variations are 2.7, 1.4, and 2.9 per cent for the three sizes of duct at the velocities given.

Substituting the value of f from Equation (17) in Equation (1) the friction loss formula for air at standard conditions and for $L = 100$ ft reduces to:

$$H = \frac{1.47\ (V/4000)^{1.83}}{D^{1.30}} \qquad (18)$$

A comparison of calculated results with test results is presented in Table 3 to check the application of this formula in calculating friction losses. The calculated results of other investigators are also shown for comparison. Column *B* shows the calculated pressure drop per 100 ft using the new formula for the velocities in Column *A*. Column *C* gives this pressure drop in per cent of the average test data values in Column *G*. It will be noted that for the three sizes of duct average per-

Table 2—Relation Between Pressure Drop in Inches of Water Per 100 Ft and Velocity for Straight Run of Ducts—No Joints

				Shop Made Duct					Factory Made		
A	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	*J*	*K*	*L*
Velocity FPM	Calculated New Lab. Eq.	Column *B* In Per Cent of *E*	Calculated—Old Lab. Eq. In Per Cent of *E*	Avg. Test Data	High Value Curve	Column *F* In Per Cent of *E*	Low Value Curve	Column *H* In Per Cent of *E*	Low Value Curve In Per Cent of *E*	Avg. Test Data In Per Cent of *E*	High Value Curve In Per Cent of *E*
4-in. Round				Includes 4 Tests					Includes 4 Tests		
400	0.0655	100.7	101.7	0.065	0.070	107.6	0.060	92.4		132.2	
800	0.234	101.7	103.7	0.230	0.240	104.3	0.218	94.7		113.8	
1200	0.490	101.2	102.7	0.485	0.485	100.0	0.485	100.0		111.0	
1600	0.820	100.0	103.6	0.820	0.840	102.4	0.820	100.0		111.2	
2000	1.235	99.6	103.1	1.24	1.265	102.1	1.24	100.0		108.8	
2400	1.735	100.3	103.3	1.73	1.768	102.1	1.73	100.0		107.2	
2800	2.30	100.0	103.0	2.30	2.36	102.5	2.30	100.0		104.3	
3200	2.93	99.6	102.6	2.94	3.02	102.5	2.94	100.0		100.3	
Avg.		100.4	102.9			102.9		98.4	Same As Avg.	111.1	Same As Avg.
8-in. Round				Includes 12 Tests					Includes 4 Tests		
400	0.0277	102.6	99.7	0.027	0.027	100.0	0.027	100.0	96.4	100.0	103.3
800	0.100	101.0	100.5	0.099	0.114	115.2	0.085	95.8	98.0	100.0	102.4
1200	0.209	100.3	99.3	0.208	0.232	111.5	0.168	80.8	100.0	101.3	102.0
1600	0.350	99.7	100.2	0.351	0.391	111.3	0.281	80.0	101.2	102.4	103.5
2000	0.525	98.8	100.3	0.533	0.590	110.8	0.442	83.1	100.4	101.3	102.2
2400	0.740	99.6	99.6	0.743	0.830	111.7	0.645	86.8	102.3	105.0	107.5
2800	0.980	99.5	100.5	0.985	1.11	112.8	0.873	88.6	100.4	103.6	106.6
3200	1.255	99.6	100.1	1.260	1.42	112.5	1.435	90.2	102.3	104.6	106.8
Avg.		100.1	100.0			110.7		88.2	100.1	102.3	104.4
24-in. Round				Includes 5 Tests							
800	0.0261	96.7	93.0	0.027	0.0275	101.7	0.029	107.5			
1200	0.0547	96.1	91.5	0.057	0.057	100.0	0.058	101.6			
1600	0.0915	96.4	93.0	0.095	0.097	102.0	0.094	98.9			
2000	0.138	98.5	95.1	0.140	0.143	102.0	0.134	95.8	No Tests Taken		
2400	0.193	99.0	95.4	0.195	0.20	102.5	0.182	93.8			
2800	0.256	100.3	97.3	0.255	0.270	105.8	0.237	93.0			
3200	0.328	102.4	98.7	0.320	0.335	104.7	0.295	92.2			
Avg.		98.5	94.9			102.7		97.5			

Notes on column headings:
(*B*) Calculated, based on new Lab. equation, $H = 1.143\ (V/4000)^{1.83}/D^{1.22}$
(*C*) Calculated, based on old Lab. equation, $H = 1.14\ (V/4000)^{1.84}/D^{1.26}$

Table 3—Relation Between Pressure Drop in Inches of Water Per 100 Ft and Velocity for Straight Run of Duct—With Joints

A	*B*	*C*	*D*	*E*	*F*	*G*
Velocity fpm	Calculated New Lab. Eq.	Column *B* In Per Cent of *G*	Calculated, Old Lab. Eq. In Per Cent of *G*	Calculated Zwickl Eq. In Per Cent of *G*	Calculated, Fan Engr. Eq. In Per Cent of *G*	Avg. Test Data
		4-in. Round				
400	0.091	102.2	99.8	141	108	0.089
800	0.326	103.6	102.3	146	109	0.315
1200	0.675	100.8	99.3	145	107	0.670
1600	1.14	100.8	100.0	147	107	1.13
2000	1.73	101.8	101.2	147	108	1.70
2400	2.41	100.4	99.6	146	107	2.40
2800	3.20	100.0	99.7	146	107	3.20
3200	4.07	100.8	100.6	149	108	4.04
Avg.		101.3	100.3	146	108	
		8-in. Round				
400	0.0368	103.8	100.0	145	113	0.0355
800	0.132	102.3	100.0	146	113	0.129
1200	0.274	101.1	99.0	147	111	0.271
1600	0.462	99.9	98.4	147	111	0.463
2000	0.699	99.5	97.8	146	110	0.703
2400	0.978	99.3	97.5	146	110	0.985
2800	1.30	99.3	99.3	147	110	1.31
3200	1.65	99.4	98.8	148	111	1.66
Avg.		100.6	98.9	146	111	
		24-in. Round				
800	0.0322	97.8	93.9	139	113	0.033
1200	0.066	95.7	92.3	140	112	0.069
1600	0.111	96.5	92.9	144	114	0.115
2000	0.166	99.0	97.0	148	119	0.168
2400	0.234	100.0	97.0	150	119	0.234
2800	0.311	101.4	98.7	152	121	0.307
3200	0.396	102.5	100.0	155	123	0.386
Avg.		99.0	96.0	147	117	

Notes on column headings:

(*B*) Calculated, based on new Laboratory equation, $H = 1.47\ (V/4000)^{1.83}/D^{1.30}$ with 40 jts./100 ft.

(*D*) Calculated, based on old Laboratory equation, $H = 1.44\ (V/4000)^{1.84}/D^{1.31}$ with 40 jts./100 ft.

(*E*) Calculated, based on Zwickl equation, $H = 2.2\ (V/4000)^{13/7}/D^{9/7}$ with unstated jt. freq.

(*F*) Calculated, based on new *Fan Engineering* equation, $H = 1.678\ (V/4000)^{1.84}/D^{1.24}$ with unstated jt. freq.

centages of 101.3, 100.6 and 99 per cent are given. The maximum variation from average test data is 3.6 per cent for the 4-in., 3.8 per cent for the 8-in. and 4.3 per cent for the 24-in. duct. Thus, it is conclusively shown that the calculated results based on the formula presented here check the average values from test data, and the amount of variation shown is negligible when all factors are taken into consideration. Column *D* gives the calculated pressure drop based on the old formula in per cent of the average test data values. It will be noted that the average of these figures is 100.3 and 98.9 per cent for the first two sizes and 96 per cent for the 24-in. duct, and the calculated results based on other sources are also given for comparison. In Column *E* are pressure losses based on the formula by Dr. Zwickl[7] in per cent of average test data, while Column *F* gives the calculated pressure drop based on the formula given in the 1938 edition of *Fan Engineering* in per cent of average test data values. It should be stated that the data of Zwickl and *Fan Engineering* are for duct of an unstated joint frequency while the Laboratory data are based on duct with 40 joints per 100 ft.

It is well to point out that the Fanning Equation (1), using the square power of the velocity, will give identical results as obtained by using the derived friction loss formulae with fractional exponents. However, at the velocity for which the pressure loss is desired, the corresponding friction factor would either have to be chosen from a curve or calculated, and this value substituted in the Fanning equation. This would necessitate extra curves and computation. The Laboratory formulae here presented, although embodying fractional exponents, eliminate friction factor and Reynolds Number entirely and offer a more complete expression for frictional resistance.

[7]The Flow of Air and Its Distribution Through Ducts, by Dr. J. R. Zwickl. (*Heating and Ventilating*, February-July, 1939).

Conclusions

1. Analysis of the test data indicates a value of 1.83 as the exponent of the velocity which fits test data best. The velocity exponent of 2 which has been generally accepted and used by many engineers does not fall within the entire range of the test data.

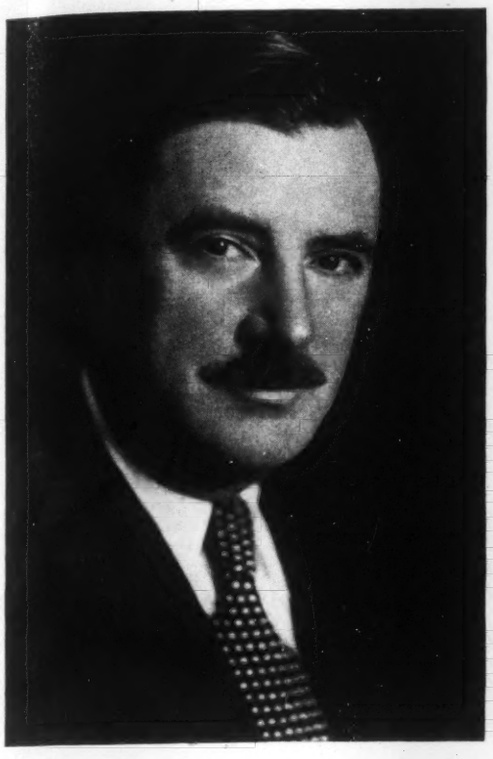
2. Calculated pressure losses, both for duct with no joints and duct with 40 joints per 100 ft, using the friction loss formulae developed in this paper, check test data to a high degree of accuracy. The close agreement between theoretical and experimental results shown for the wide range of velocities and duct sizes studied serves to lend confidence to the acceptance of these formulae in calculating pressure losses for smaller, intermediate, and larger sizes of duct.

Hearing Tests Analyzed at World's Fairs

A considerable percentage of the visitors to the Bell System exhibits at both the New York Fair and the Golden Gate Exposition at San Francisco have taken advantage of the opportunity to find out how well they hear and have thus made possible the widest survey of hearing that has ever been made. Information available heretofore in the Bell Laboratories has been based on studies of a few thousand people at most, while the largest previous survey, that of the United States Public Service in 1936, covered only about 9,000. Some three-quarters of a million records are available from the World's Fairs' data. The Bell System deals in hearing and if it is to provide the best service at the lowest possible cost, it is important that it have available reliable data on the hearing characteristics of the American people.

The evaluation of the effect on one's ability to hear in any particular band of frequencies is a function of many factors. It has been found, however, that one's ability to understand speech can be determined from the average of his hearing losses at 440, 880 and 1760 cycles as compared to good young ears. If this average is 25 db, there may be some difficulty in hearing in auditoriums and churches, while if it is 45 db, there may be difficulty in hearing in direct conversation. Only if the hearing loss is as much as 65 db will there ordinarily be great difficulty in hearing over the telephone. The tests indicate that about 1 out of 25 persons have difficulty in hearing in auditoriums; 1 in every 125 will have some difficulty in direct conversations; and 1 in 400 over the telephone.

During part of the summer, visitors at each Fair were asked to indicate whether they lived in the vicinity or not, so that information would be available to determine sectional differences in hearing acuity. Studies of these answers, however, do not reveal any significant difference between New York and San Francisco. The time of day at which each test is taken is marked on the card by the attendant, thus permitting a comparison of the results for different periods of the day. No consistent difference has been found, however, even between early morning and late afternoon. There is thus no indication of any effect of fatigue on hearing. The results of the tests have fortified the Bell System with additional information for the building of a progressively more satisfactory telephone system.



J. F. McIntire, Detroit, Mich.
President—1939

Proceedings of the 46th Annual Meeting

F. E. Giesecke, College Station, Tex.
President—1940

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

Cleveland, Ohio, January 22-26, 1940

WITH an attendance that approached an all-time record the 46th Annual Meeting of the Society was held in the Hotel Statler, Cleveland, Ohio, January 22 to 26. In a total registration of over 1,100 members, ladies and guests, it was notable that over one-fifth of the total Society membership was registered at this meeting. The members were attracted by an important group of committee meetings, a fine technical program, and the 6th International Heating and Ventilating Exposition. The turnout was such that the city's hotel facilities were taxed to the limit.

FIRST SESSION—

Tuesday, January 23, 9:30 A.M.

The 46th Annual Meeting was called to order by Pres. J. F. McIntire in the grand ballroom of the Hotel Statler, Cleveland, Tuesday morning, January 23. Hon. Harold L. Burton, mayor of the City of Cleveland, welcomed the members and said, "We appreciate your coming as it enables our engineers in this part of the country to share the benefits that you bring in advancing the interests of heating, ventilating, and air conditioning."

He also said, "In these difficult times of the depression, I have often thought that although the times are hard and although we go through great difficulties of one kind or another, there are a great many things that are helpful to us, and I believe that a part of that help is in the pressure that is being put upon us to develop new mechanical and scientific appliances and devices."

He said further, "There are so many more foci of development in our generation than there ever were before; there are so many more places to develop the man-made wealth than there ever have been before, that there is a guarantee that we will develop it. There is much more knowledge available because no longer does a man invent by himself, but he has the wealth of the world; the translated knowledge of the world is his research bureau to help him about it. An inventor is no longer a man unto himself, but a man with the help of all the world at his hand to help him do the inventing and advancing. The means of obtaining materials and distributing products are so much more improved that, personally, I believe if we will but let ourselves use them, this lag between new invention and unemployment that follows can be cut down so that we will in our generation be able to move up the advance from one invention to another to a much closer period and thereby find in it a much greater amount of wealth available to us than before.

"I want to illustrate my point by the radio which, of course, 20 years ago was unknown. Today it is with us with 340,000 employees in that industry where there were none 20 years ago. That means a million and a half people dependent upon a new mechanical device and scientific advance.

"The same thing is true in your own field. You are moving into a field that has never been fully developed or properly understood in the nation. It is a field, therefore, in which we have an opportunity by realizing that

by the combination of our efforts and our knowledge we are able to move into a field of wealth, a standard of living that has not been available to us before.

"And so as you come here we welcome this industry that is advancing into that new field as you are. We welcome you as a city that has been largely publicized in the past few months because of its relief troubles. That does not mean that the city is suffering an industrial collapse at all. It is just an indication of the psychology that sometimes wrongly follows a bit of publicity.

"We are moving ahead here. We are feeling the effect of an advance in industry, of an advance in new products and we welcome here an industry that will help us to look into the future and create needed added wealth that can be added to our standard of living.

"I hope that you enjoy your stay here with us. We enjoy having you with us. We want to do everything possible to make your stay not only pleasant but productive to you."

President McIntire responded briefly and expressed the appreciation of the Society for the cordial welcome given.

In a brief resumé of his administration he pointed out that the accomplishments of the past 12 months would not have been possible without the work that had been done in previous years, so that the progress of the Society was cumulative. He paid tribute to the work of the Guide Publication Committee in producing an outstanding volume that contains more technical data in a smaller book. He mentioned the progress of the Society's research work and the excellent financial condition of the Society. He complimented the Meetings Committee for the fine programs that had been arranged for the general meetings of the Society, and stated that the Standards Committee was making progress in establishing a logical procedure for the development and administration of codes. President McIntire also mentioned the excellent work being done by the chapters and the value of the Chapter Delegates' Conference held at each annual meeting, and in conclusion he expressed appreciation to past presidents for their cooperation and assistance during his administration.

The report of the secretary was given by A. V. Hutchinson as follows:

Report of Secretary

During 1939 the Society gained much in prestige and it has definitely extended its sphere of influence professionally as well as geographically. With a current membership of 3147, and local chapters in 29 principal cities, it requires a New York headquarters office staff of 10, and a research staff of 19 in the Laboratory at Pittsburgh, as well as the cooperative efforts of several hundred members serving on committees and in chapters to carry on the activities sponsored by the Society.

A brief review of the work at the headquarters office will give some idea of the various duties required to administer the financial, membership, publication, meetings, employment, chapters and committees.

In the 300 working days of the calendar year it is necessary to carry on certain routine matters specified by the Constitution and By-Laws, as well as various special projects sponsored by the Council and Committees.

Some of the accomplishments of the past year may be mentioned as follows:

1. Collection and deposit of dues, research pledges, publication and other miscellaneous income and disbursements of budgeted expenditures of $97,650.00 for general Society accounts and $27,936.00 for research. Sending quarterly dues bills and monthly statements on miscellaneous accounts.

2. Keeping the Roll of Membership up-to-date; advising chapters and Journal publishers of changes of addresses. Carrying on the procedure of membership, elections totalling 388, and cancellations, as directed by the Council.

3. Securing papers, editing manuscripts, preparing notes and discussions for the monthly JOURNAL and TRANSACTIONS. Handling the sales promotion campaigns for the sale and distribution of 11,000 copies of THE GUIDE 1939, and sale of catalog data space for 1940 edition, as well as styling and coordinating and indexing 46 chapters of the Text Section.

4. For the first time three general meetings of the Society were held during one year, respectively, in Pittsburgh, Pa., Mackinac Island, Mich., and Atlanta, Ga. This plan permitted the attendance of a great many members from different sections of the country, and the result of holding three meetings was most gratifying because the total registration of members and guests was close to 1000.

5. Employment file was maintained for the benefit of members and a great many placements were recorded during the year.

In the membership service work assistance was given to the Local Chapters by the Speakers Bureau Committee and 15 assignments were completed and, in addition to making the preliminary arrangements, advance publicity material, and other essential correspondence was required in this activity. Assistance was given by the Secretary's office in the organization of three new Chapters during the year, namely, North Carolina, Oregon and Delta Chapter at New Orleans.

It should be noted in the report of the Finance Committee that the cost of administering membership activities amounted to $1.25 per member in excess of income, but a surplus from Guide operations permitted an increase in the General Fund of the Society for the year 1939, so that it is in excellent financial condition and membership is gaining in strength.

Respectfully yours,

A. V. HUTCHINSON, *Secretary.*

A. J. Offner, New York, chairman of the Finance Committee read his report which summarized the audit of the Society by the Certified Public Accountant.

Report of Finance Committee

In submitting the financial report for the calendar year 1939, your Finance Committee is pleased to advise that the finances of the Society continue in an excellent and sound condition.

The accounts of the Society have been audited by a Certified Public Accountant as required by the By-Laws. The Balance Sheet shows that the net assets of the Society, including Research, amount to $112,923.38. Of this total there is $66,081.54 in the operating and surplus accounts and $30,596.80 in the Reserve and Endowment Funds and invested in securities of the United States Government. Comparing the total income and expenditures of the Society during the past year with the estimated figures set up in the Budget as adopted by the Council at the beginning of the year, it is found that the income was about $5,000 under estimates, while expenses were nearly $14,000 less than Budget provisions. The operations of general Society activities show a loss for the year, but with the surplus from GUIDE operations and deductions of reasonable reserves leaves an addition to the general fund amounting to $1,642.87. Research activities had an excess of $7,626.38 in expenditures over current income, the difference being made up by reserve funds.

The detailed statement of the Society's assets and liabilities as prepared by the Society's Certified Public Accountants, Tusa and La Bella, is attached to this report and will be published in the Society's JOURNAL.

Subject to the requirements of the Constitution, 40 per cent of the dues collected from Members and Associates amounting to $14,903.73 was turned over to the Committee on Research.

In addition to this amount the Council appropriated $5,000, making a total of $19,903.73 contributed by the Society during the past year for Research work. The total amount spent for Research in 1939 was $44,262.39, which in addition to Society contributions includes earmarked contributions, unearmarked contributions and contracts. A detailed financial statement of the Committee on Research will be presented by the Chairman of that Committee.

As a part of its duties, and with the approval of the Council, the Finance Committee sold above par $4,000 par value U. S. Treasury Bonds due in 1941. This sale was made to protect the financial interest of the Society due to the nearness of the maturity date and the general tendency for such bonds to gradually drop to par. The present market value for this series of Treasury Bonds has proven the soundness of disposing of them. The Constitution and By-Laws of the Society limits the investment of surplus Society funds to U. S. Government Securities and those securities which are legal for savings banks in the State of New York. During the past year, the Finance Committee, with Council approval, has invested $7,500, the limit permitted by law, in U. S. Savings Bonds of $10,000 par value at maturity date. This will give an interest return at maturity of 50 per cent more than could have been obtained had this money remained in savings banks accounts. These bonds can be sold at any time at the original cost plus interest to date of sale. Any further surplus funds available will be invested at opportune time for sound and legal investments. Pending this opportunity your Committee recommends holding a cash reserve in savings banks accounts.

The Constitution and By-Laws require that the general accounts and the Research accounts of the Society be segregated and be kept separate. During the past year, the books and accounts formerly kept at Pittsburgh have been transferred to New York and are now being handled by the office staff in New York.

Respectfully submitted,

A. J. OFFNER, *Chairman*
Finance Committee.

Report of Certified Public Accountant

January 18, 1940.

American Society of Heating and
Ventilating Engineers,
51 Madison Avenue,
New York, N. Y.

Gentlemen:

Pursuant to your request, we made an examination of the books of account and records of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—New York, N. Y., and the Research Fund for the calendar year ended December 31, 1939, and submit herewith our report.

The work covered a verification of the assets and liabilities as of the date previously stated and a review of the operating accounts for the year ended December 31, 1939. For the period recorded cash receipts were traced into the depositories; the cancelled bank vouchers were inspected and compared with the cash records and the dues income was accounted for.

Submitted herewith is a Balance Sheet showing the financial condition of the Society on December 31, 1939, and your attention is directed to the following comments thereon:

CASH

Cash on Deposit was verified by direct communication with the banks and reconcilement of the amounts reported to us with the balance shown by the books of the Society.

The Petty Cash kept in the New York Office was verified by count.

MARKETABLE SECURITIES

There is attached hereto a schedule of negotiable bonds which were verified by direct communication with the Bankers Trust Company where same are deposited for safe-keeping. No adjustment has been made of the $3,280.21 shrinkage in the market value of these securities, same being included in the attached Balance Sheet at cost.

ACCOUNTS RECEIVABLE

A list of the Membership Dues Receivable as of December 31, 1939, furnished to us by the management was checked to the individual ledger cards and found in agreement with the General Ledger Control. The unpaid dues may be summarized as follows:

Dues billed in 1939	$ 9,861.46
Dues billed in 1938	2,961.59
Dues billed in prior years	4,643.44
TOTAL	$17,466.49

Amounts due from GUIDE Advertisers and other Accounts Receivable were verified by trial balance of the individual ledger; accounts were found in agreement with the General Ledger Control.

After charging off $3,939.44 in uncollectible dues per council authorization we have increased the Reserve for Dues Doubtful of Collection $7,630.28. The adjusted reserves now shown on the attached Balance Sheet in our opinion are ample to cover any losses from future realization.

INVENTORIES

Items appearing under this caption are based upon quantities submitted by the management and computations made by us. The Transactions inventory priced at cost may be summarized as follows:

Prior years	$1,750.00
1933-1934	670.46
1935-1936	75.90
1937-1938	448.72
TOTAL	$2,945.08

FURNITURE, FIXTURES AND LIBRARY

Furniture, Fixtures and Library are shown herein at the book values without appraisal by us. We did, however, provide for Depreciation of Furniture and Fixtures at the rate of ten per cent (10%) per annum.

ACCOUNTS PAYABLE

On December 31, 1939, there remained unpaid invoices amounting to $9,976.00 which included the sum of $9,407.42 estimated by the management necessary to complete and make the first mailing of the 1940 GUIDE.

PAYABLE TO RESEARCH FUND

The amounts due the Research Fund, representing 40% of Members' and Associates' dues, were determined from computations made by us in accordance with Section 5, Article 3, of the By-Laws.

ACCRUED ACCOUNTS

Additional compensation to the secretary and the clerical staff of the Society has been computed in accordance with the instructions of the Finance Committee.

DEFERRED INCOME

Members' prepaid dues were ascertained by trial balance of the individual ledger cards. Prepaid dues from proposed mem-

bers were verified from inspection of the applications found on file.

GENERAL FUND

There is attached hereto a schedule showing the changes that have occurred in the General Fund during the calendar year 1939.

SPECIFIC FUNDS

Specific Funds are shown in the attached Balance Sheet after adjustment for interest earned.

The Balance Sheet and schedules showing the operations of the Research fund are annexed hereto, also a separate report has been prepared and rendered to the Committee on Research.

OPERATIONS

There are attached hereto Statements of Income and Expenses for the calendar year 1939 showing an excess of Expenses Over Income from Society activities of $3,959.52 and a net income of $5,602.39 from GUIDE Operations. In the preparation of these statements 30% of the salaries and office expenses have been allocated to GUIDE Operations.

Respectfully submitted,

TUSA & LA BELLA,
Certified Public Accountants.

BALANCE SHEET

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
New York, N. Y.
December 31, 1939

ASSETS

SOCIETY			
CASH			
On deposit		$36,675.12	
On hand		4.03	
In closed banks		418.97	$ 37,098.12
RESERVE FUNDS			
On deposit			$ 4,294.29
INVESTMENTS (AT COST)			
Securities (Market Value $18,919.17)		$18,678.23	
Add: Accrued interest		4.50	$ 18,682.73
ACCOUNTS RECEIVABLE			
Membership dues	$17,466.49		
Less: Reserve for doubtful	11,796.69	$ 5,669.80	
Advertisers and sundry debtors	$31,098.81		
Less: Reserve for doubtful	1,462.62	$29,636.19	$35,305.99
INVENTORIES			
Transactions—Copies		$ 2,945.08	
Transactions—Paper		371.28	
Emblems and certificate frames		183.35	
Stationery		116.38	$ 3,616.09
PERMANENT			
Library		300.00	
Furniture and fixtures	$ 7,209.69		
Less: Reserve for depreciation	5,447.56	1,762.13	2,062.13
			$101,059.35
SPECIFIC FUNDS			
ENDOWMENT FUND			
Cash on deposit		$13,828.19	
Securities at cost (market value $8,397.50)	$11,918.65		
Add: Accrued interest	111.35	$12,030.00	$ 25,858.19
F. PAUL ANDERSON AWARD FUND			
Cash on deposit			1,126.01
			$128,043.55

LIABILITIES AND GENERAL FUND

SOCIETY			
ACCOUNTS PAYABLE			$ 9,976.00
PAYABLE TO RESEARCH FUND			
On dues as and when collected			6,342.61
ACCRUED ACCOUNTS			
Compensation—Secretary and Staff			2,614.21
RESERVE FOR PUBLICATION			
Transactions—Volume 45			3,700.00
DEFERRED INCOME			
Prepaid dues—Members	$ 346.42		
Prepaid dues—Prepaid members	316.63	$ 663.05	
TOTAL LIABILITIES			$23,296.87
GENERAL FUND			
Society			$77,756.48
TOTAL LIABILITIES AND GENERAL FUND			$101,059.35
SPECIFIC FUNDS			
ENDOWMENT FUND			
Principal		$24,063.28	
Unexpired income		1,794.91	$25,858.19
F. PAUL ANDERSON AWARD FUND			
Principal		$ 1,000.00	
Unexpired income		126.01	$ 1,126.01
			$128,043.55

NOTE "A": This Balance Sheet is subject to the comments contained in the letter attached to and forming a part of this report.

BUDGET COMPARISON

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
New York, N. Y.

For the Calendar Year Ended December 31, 1939

	Actual	Budget Provision	Increase Decrease
Income:			
Initiation Fees	$ 2,547.00	$ 1,500.00	$ 1,047.00
Dues Collectible	24,318.09	26,000.00	1,681.91
Editorial Contract	15,648.31	15,000.00	648.31
Profit—Emblems and certificate frames	98.12	100.00	1.88
Profit—Reprints and books	232.55	300.00	532.55
Sales of TRANSACTIONS	1,077.73	750.00	327.73
Interest—Securities and savings accounts	1,213.93	1,000.00	213.93
GUIDE advertising	27,440.10	27,000.00	440.10
GUIDE sales	18,966.97	26,000.00	7,033.03
	$91,077.70	$97,650.00	$ 6,572.30
Expenses:			
General Society Activities:			
Meetings	$ 2,727.77	$ 1,800.00	$ 927.77
Chapter allowance	677.94	1,000.00	322.06
Promotional expense	589.71	1,000.00	410.29
Speakers Bureau	1,209.02	1,500.00	290.98
Subscriptions—HPAC	6,308.23	6,300.00	8.23
Postage	1,712.19	2,000.00	287.81
Rent and light	3,601.99	4,000.00	398.01
TRANSACTIONS	4,467.79	3,700.00	767.79
Codes		1,000.00	1,000.00
Salaries—Secretary and staff	17,731.00	18,100.00	369.00
General printing	499.35	700.00	200.65
Membership certificates and emblems	185.10	600.00	414.90
President's fund	1,100.20	1,800.00	699.80
Secretary travel	1,450.03	1,200.00	250.03
Council and Chapter Delegates travel	3,413.75	4,000.00	586.25
Bank charges	103.36	100.00	3.36
Multigraphing	280.63	600.00	319.37
Telephone	661.94	700.00	38.06
Telegraph	275.30	350.00	74.70
Professional services	500.00	650.00	150.00
General office expenses	555.58	650.00	94.42
Office supplies	659.49	600.00	59.49
Furniture and fixtures	240.97	500.00	259.03
Reserve for additional compensation	2,614.21	2,100.00	514.21
Special appropriations	5,268.12	1,200.00	4,068.12
	$56,833.67	$56,150.00	$ 683.67

Cost of GUIDE:			
Copy sales promotion (1939 edition)	$ 3,285.90	$ 3,500.00	$ 214.10
Expressage and mailing (1939 edition)	3,028.54	4,500.00	1,471.46
Advertising sales promotion	1,452.32	3,000.00	1,547.68
Editorial and advertising salaries	7,000.00	7,500.00	500.00
Committee expense	441.89	500.00	58.11
Printing 1940 edition, 13M copies	9,008.62	12,000.00	2,991.38
Binding 1940 edition	4,013.25	4,500.00	486.75
Paper 1940 edition	2,879.08	4,000.00	1,120.92
Cuts and drawings	429.27	500.00	70.73
Yearbook	1,062.29	1,500.00	437.71
	$32,601.16	$41,500.00	$ 8,898.84
	$89,434.83	$97,650.00	$ 8,215.17

BALANCE SHEET

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
Research Fund—New York, N. Y.

December 31, 1939

ASSETS

CASH			
ON DEPOSIT			
Treasurer's account — Bankers Trust Co.	$2,013.28		
Secretary's account — Chase National Bank	1,205.53		
Director's account — Forbes National Bank	77.96		
Thrift account—Bank for Savings	3,148.39	$6,445.16	
ON HAND FOR DEPOSIT			
Treasurer's account — Bankers Trust Co.	3,050.60		
Director's account — Forbes National Bank	224.95	3,275.55	
ON HAND			
Petty cash—Pittsburgh		16.29	$ 9,737.00
ACCOUNTS RECEIVABLE			
RESEARCH CONTRACTS			
Navy Department		2,081.09	
Pittsburgh Corning Corp.		250.00	
Owens Illinois Glass Co.		250.00	2,581.09
PERMANENT			
Laboratory equipment			1.00
Total Research Fund assets			$12,319.09
FUND:			
RESEARCH ENDOWMENT FUND			
CASH ON DEPOSIT			
Bank for Savings			697.93
			$13,017.02

LIABILITIES AND FUND

ACCOUNTS PAYABLE			
Trade creditors	$ 326.40		
DEFERRED INCOME			
Radiation and comfort study	$2,017.69		
Air conditioning requirements of glass	846.83		
Corrosion study	500.00		
Convector study	400.00		
Air distribution	750.00	4,514.52	$4,840.92
RESEARCH FUND		7,478.17	
Total Research Fund liabilities and fund			$12,319.09
FUND			
RESEARCH ENDOWMENT FUND			
Principal		600.00	
Unexpired income		97.93	697.93
			$13,017.02

NOTE "A"—This balance sheet is subject to the comments contained in the letter attached to and forming a part of this report.

NOTE "B"—The Research Fund as at December 31, 1939, had a contingent asset amounting to $6,342.61 arising from 40% of the Society's members and associates dues, payable upon collection by the latter.

BUDGET COMPARISON—RESEARCH FUND

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
New York, N. Y.

(For the Calendar Year Ended December 31, 1939)

	Actual	Budget Provision	Increases Decreases*
Balance from 1938 committee—Cash in Pittsburgh and New York banks	"A" $15,253.55		$15,253.55*
Income:			
Budgeted:			
Dues—Current and prior years	$14,903.73	16,000.00	1,096.27
Special council appropriations	5,000.00	5,000.00	.....
Interest on savings accounts	62.17	60.00	2.17
Contributions—General research	2,425.00	2,000.00	425.00
Contributions—Earmarked Research —radiation and comfort	3,982.31	6,000.00	2,017.69
Contributions—Earmarked —glass study	1,562.80	"A" 2,409.63	846.83
	$27,936.01	$46,723.18	$18,787.17
Unbudgeted:			
Earmarked research—Navy contract	$ 5,000.00	.....	$ 5,000.00
Earmarked research—Glass block	3,500.00	.....	3,500.00
Earmarked research—Cooling load	200.00	.....	200.00
	$36,636.01	$46,723.18	$10,087.17
Expenditure:			
Budgeted:			
Salaries—Laboratory staff	$ 9,419.60	$13,500.00	$ 4,080.40
Salaries—Student help	477.50	1,800.00	1,322.50
Salaries—Correlation thermal research	1,500.00	1,500.00	.....
Salaries—Office of chairman	300.00	300.00	.....
Travel—Committee and laboratory staff	1,647.93	3,000.00	1,352.97
Laboratory supplies and equipment	873.98	1,000.00	126.02
Office supplies and equipment	781.47	600.00	181.47
Printing	159.39	250.00	90.61
Meetings	213.16	300.00	86.84
Exhibits	226.40	250.00	23.60
Promotion and publicity	222.67	500.00	277.33
Contract—U. S. Bureau of Mines	101.83	500.00	398.17
Cooperative research	8,525.00	10,550.00	2,025.00
Earmarked research—radiation and comfort	3,982.31	6,000.00	2,017.69
Earmarked research—Glass study	1,562.80	2,409.63	846.83
Earmarked research—Cooling load	4,345.74	600.00	3,745.74
Earmarked research—Air conditioning	1,147.25	2,500.00	1,352.75
Contingencies	165.47	500.00	334.53
	$35,651.60	$46,059.63	$10,408.03
Unbudgeted:			
Earmarked research—Navy contract	$ 5,000.00	.....	$ 5,000.00
Earmarked research—Glass block	3,610.79	.....	3,610.79
	$44,262.39	$46,059.63	$ 1,797.24
Balance	7,626.38	663.55	8,289.93
	$36,636.01	$46,723.18	$10,087.17

The report of the Council was given by the secretary, as follows:

Report of Council

To administer the business of the Society the Council held five meetings during the year just completed. The organization meeting was held January 26, 1939 at Pittsburgh, Pa., with President McIntire presiding. He announced the appointment of five Council Committees and five Special Committees, and the Chairman and Vice-Chairman of the Committee on Research respectively, Messrs. W. L. Fleisher and J. H. Walker.

Additional special committees were appointed during the year as needed.

Appointment of the Secretary and Technical Secretary was made and depositories for Society funds were selected. The Budget for 1939 provided for an estimated income of $97,650.00 and an expenditure of $97,650.00. The Research Budget amount-

ing to $27,936.00 was approved, and $5,000 was appropriated from Society funds for research work.

The accounting firm of Tusa and LaBella, Certified Public Accountants, was selected to audit Society accounts and rendered two reports during the year.

During the summer, two meetings of the Council were held at Mackinac Island, July 4 and 6, at which reports of the Special Committees were received. Approval was given to the investment of $7,500 in United States Savings Bonds (legal limit permitted by law).

A Charter was granted for the organization of a Local Chapter in Portland, Ore., upon petition of 36 members. Nominees for a three-year term on the Committee on Research commencing in 1940 were made.

The Council inaugurated a new plan and authorized a fall meeting of the Society in Atlanta, Ga., October 30 and 31.

Washington, D. C. was selected as the meeting place for the summer meeting 1940.

The fourth meeting of the Council in 1939 was held in Atlanta, Ga., October 29, and reports were received from regular and special committees. Provision was made for payment of chapter delegates' transportation expenses to the 46th Annual Meeting.

A Committee was appointed to review the Society's publication policy and negotiated a new editorial contract. The Exchange Service Plan with the *ASRE* was approved for 1940.

Legal advice was sought on the status of members who had given notes and agreements for deferred payment of Society's dues several years ago.

During the year routine action was taken in providing for election of members to the Society, acceptance of resignations, cancellations as required by the Constitution and By-Laws.

Petition for the 29th Chapter of the Society from New Orleans members was received and voted upon by letter ballot and a Charter was granted for the Delta Chapter of the ASHVE with headquarters in New Orleans.

On January 22 the final meeting of the Council was held in Cleveland, Ohio, and received reports of committees and other unfinished business.

Respectfully submitted,
A. V. Hutchinson, *Secretary.*

President McIntire called upon Prof. A. B. Algren, Minneapolis, to present the first technical paper entitled Effect of Lint on Air Filter Performance by Frank B. Rowley and Richard C. Jordan (complete paper published in January 1940 ASHVE Journal Section, *Heating, Piping and Air Conditioning*). C. F. Mally, Detroit, inquired about the usual life of a filter, and A. J. Rummell, San Antonio, Tex., asked about the proper percentage of lint found in air in residences and office buildings. G. E. May, New Orleans, asked about the method of dust feeding during the test, and P. B. Elliott, Rochester, inquired as to the method used to determine whether lint passed through the filter.

The second paper entitled Gas Equipment for Year-Round Air Conditioning, by G. E. May (complete paper to be published in ASHVE Journal Section, *Heating, Piping and Air Conditioning*), was given by the author. He was asked questions on the value of an economic analysis for each installation by A. D. Marston, Kansas City, and H. M. Hart, Chicago, inquired about limiting applications to extended surface type of coils. Comments were also offered by E. K. Campbell, Kansas City, on the use of pump for circulation of water for heating, and D. S. Cooper, Houston, on humidity control. A. J. Rummell, San Antonio, asked about the percentage of total operating costs represented by electric auxiliary equipment, and J. N. Hadjisky, Detroit, inquired about humidity control with large latent heat load.

President McIntire then presented the Report of the Tellers as follows:

Report of Tellers of Election

Ballot—Officers

	For
President—F. E. Giesecke	634
First Vice-President—W. L. Fleisher	619
Second Vice-President—E. Q. Eastwood	633
Treasurer—M. F. Blankin	633
Members of the Council—Three-Year Term	
J. F. Collins, Jr.	634
E. N. McDonnell	633
T. H. Urdahl	634
C.-E. A. Winslow	634
Total Ballots Counted	698
Ballots Invalid	64

Ballot—Committee on Research

Three-Year Term	
Philip Drinker	633
Axel Marin	633
A. E. Stacey, Jr.	624
J. H. Van Alsburg	630
J. H. Walker	634
Total Ballots Counted	698
Ballots Invalid	64

Board of Tellers,
C. H. Quirk, *Chairman.*
W. W. Timmis.
H. S. Wheller.

The concluding paper at the morning session was given by H. B. Nottage, Chicago, entitled Dynamic and Thermal Behavior of Water Drops in Evaporative Cooling Processes, by Mr. Nottage and L. M. K. Boelter (complete paper to be published in ASHVE Journal Section, *Heating, Piping and Air Conditioning*). For the discussion 1st Vice-Pres. F. E. Giesecke took the chair. S. R. Lewis, Chicago, complimented the authors for the fundamental information developed and stated that it would be invaluable to designing engineers. C. H. Chalmers, Minneapolis, W. H. Driscoll, New York, Mr. Moore, Washington, and W. L. Fleisher, New York, commented on the value of the paper, and thought that the cooperative research carried on at the University of California would continue to develop some important engineering data.

At the conclusion of the discussion L. T. Avery, chairman of the Committee on Arrangements for the Northern Ohio Chapter was introduced and welcomed the members and guests to Cleveland on behalf of the Chapter, and outlined some of the entertainment features.

SECOND SESSION—

Tuesday, January 23, 2:00 P.M.

The second technical session convened at 2:00 p. m. with 1st Vice-President Giesecke presiding. The first paper entitled Heat Gain Through Glass Blocks by Solar Radiation and Transmittance, by F. C. Houghten, David Shore, Harold T. Olson and Burt Gunst (complete paper to be published in ASHVE Journal Section,

Heating, Piping and Air Conditioning) was presented by Mr. Houghten. Comments were offered by H. B. Nottage on the characteristics and shapes of glass block surfaces which would affect heat flow. C. H. Randolph, Milwaukee, asked about the effect of Venetian blinds, and W. C. Randall, Detroit, inquired as to surface temperatures on various types of glass surfaces.

Prof. E. R. Queer, State College, Pa., presented the paper entitled Thermal Test Coefficients of Aluminum Insulation for Buildings, by G. B. Wilkes, F. G. Hechler and Professor Queer (complete paper published in January 1940 ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*). Prepared discussion by P. F. McDermott, Manville, N. J., was read. R. D. Watt, Seattle, inquired about the effect of insulation practices on the heat transmission coefficient and Mr. Nottage suggested a method of dividing over-all coefficients into three divisions, convection, radiation and conduction. Professor Wilkes replied to these questions, and J. D. Edwards, New Kensington, Pa., commented on the effect of applying lacquer to aluminum foil surface.

The next paper entitled Diminishing Effectiveness of Successive Thicknesses of Insulating Materials by Paul D. Close (complete paper published in March 1940 JOURNAL SECTION, *Heating, Piping and Air Conditioning*) was presented by the author. Written discussion was given by P. F. McDermott, Manville, N. J., and Wharton Clay, New York, pointed out the reasons which he believed demonstrated the futility of selecting insulation on the basis of the law of diminishing returns. R. K. Thulman, Washington, D. C., said that the Federal Housing Administration was concerned with the relation of maximum heat loss and the economics of heating within certain budget limitations, and that the return on the investment for insulation should be allowed on an engineering basis so that the return on the investment would occur within a specified period of time. Comments were offered by Prof. W. T. Miller, Lafayette, Ind., Prof. A. B. Algren, Minneapolis, Prof. W. M. Sawdon, Ithaca, on the cost of insulation, and immunity from moisture troubles of structural types, and the placement of insulation and its relation to heat saving. A. T. Miller, New York, thought that the author's approach was not sufficiently comprehensive. The author answered the various questions at the conclusion of the discussion.

President McIntire announced the final paper of the afternoon session, Calculation of Coil Surface Areas for Air Cooling and Dehumidification, by John McElgin and D. C. Wiley (complete paper published in March 1940 ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*), which was presented by Mr. Wiley. Prof. G. L. Tuve, Cleveland, commended the authors for clarifying a complex heat transfer problem, and he presented another method of solving the problem. H. B. Nottage, referred to the work reported by London, Mason and Boelter to the *ASME* on the performance characteristics of a mechanically induced draft contraflow packed cooling tower. S. H. Downs, Kalamazoo, asked about the effect of a heavy water film on tube surfaces and also the effect of ice and frost. Replies to the various questions were made by Mr. McElgin.

THIRD SESSION—

Wednesday, January 24, 9:30 A.M.

President McIntire called the third session to order on Wednesday morning, January 24, and announced the Report of the Committee on Research presented by W. L. Fleisher, chairman of Committee on Research (complete report published in January 1940 ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*).

The first technical paper entitled The Peripheral Type of Circulatory Failure in Experimental Heat Exhaustion —The Role of Posture, by R. W. Keeton, F. K. Hick, Nathaniel Glickman and M. M. Montgomery (complete paper published in February 1940 ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*) was presented by Dr. Keeton.

For the discussion on this paper 2nd Vice-President Fleisher took the chair. Dr. C.-E. A. Winslow, New Haven, expressed great interest in the paper, and John Howatt, Chicago, complimented Dr. Keeton on his excellent presentation. Chairman Fleisher called upon A. E. Stacey, Jr., New York, new chairman of the Committee on Research, who brought up the question of what happens to a person if he gets down to the exhaustion point and is brought out into proper conditioning. W. R. Rhoton, Cleveland, inquired whether the increase of air motion with the increase in the relative humidity overcomes the effects on the subjects. Dr. Hick replied to the various questions asked during the discussion.

The next paper presented, after 1st Vice-President Giesecke took the chair, Advantages of Bactericidal Ultraviolet Radiation in Air Conditioning Systems, by H. C. Rentschler and Rudolph Nagy, (complete paper published in February 1940 ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*) was given by Dr. Rentschler. John Howatt, Chicago, inquired about the limits of wavelength and frequency of vibration that is effective, and C. F. Eveleth, Cleveland, questioned the description of the system as recirculating and asked if no outside air at all was taken. Dr. Winslow, New Haven, and J. J. Aeberly, Chicago, presented interesting phases of the subject. Further comments were offered by W. L. Fleisher and H. M. Hart, and Dr. Rentschler responded to the questions raised.

Mr. Fleisher again took the chair, and announced the next technical paper, Seasonal Variation in Reactions to Hot Atmospheres by F. C. Houghten, A. A. Rosenberg and M. B. Ferderber (complete paper published in February 1940 ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*), which was given by Mr. Houghten. Dr. Winslow commended the authors for this contribution, and mentioned that one may conclude from it that a given temperature maintained indoors in winter is considerably worse than the same thing in summer.

FOURTH SESSION—

Wednesday, January 24, 2:00 P.M.

A joint session of the members of the *National Warm Air Heating and Air Conditioning Association* and the ASHVE was held at the Hollenden Hotel at 2:00 p. m.

on Wednesday, January 24. Pres. L. R. Taylor, *NWAHACA*, called the meeting to order and presented the gavel to J. F. McIntire, president of the ASHVE. He then introduced W. L. Fleisher, chairman of the ASHVE Committee on Research, who gave a brief outline of the program being carried on by the Society. He pointed out some of the practical applications of research results which have been developed since the establishment of the Society's Research Laboratory 20 years ago. The National Warm Air Research sponsored by the Association was outlined by F. L. Meyer in a brief talk.

An interesting address was delivered by C. H. Mylander, banker, Columbus, Ohio.

A research paper entitled Performance of a Stoker-Fired Warm-Air Furnace as Affected by Burning Rate and Feed Rate, by A. P. Kratz and S. Konzo (complete paper published in January 1940 ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*), was given by Professor Konzo. He used stereopticon slides and a movie film to show some of the results obtained in investigations carried on at the Research Residence of the *NWAHACA* at the University of Illinois.

FIFTH SESSION—

Thursday, January 25, 10:00 A.M.

The last technical session convened at 10:00 a. m. on Thursday, January 25, with President McIntire presiding. He introduced T. M. Dugan, McKeesport, Pa., who represented the ASHVE on Committee B-16 of the *American Standards Association*, and who submitted a written report which he outlined briefly at this session.

The first technical paper at this final session entitled Analysis of the Factors Affecting Duct Friction, by J. B. Schmieler, F. C. Houghten and Harold T. Olson (complete paper published in March 1940 ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*), was presented by Mr. Schmieler. J. H. Van Alsburg, Chicago, gave brief comments on the paper.

There was an intermission from the technical session and President McIntire discussed the progress of the membership in the Society during the past year, and brought out the fact that he believed a steady growth was much better for the Society than membership drives that would bring in members who would stay for a year and drop out. He mentioned those members who were present at the meeting who joined the Society 25 years ago, and called upon W. H. Driscoll, past president of the Society, to escort the incoming president to the chair. The newly elected officers who were inducted into office were: Pres. F. E. Giesecke, College Station, Tex., 1st Vice-Pres. W. L. Fleisher, New York, 2nd Vice-Pres. E. O. Eastwood, Seattle, Treas. M. F. Blankin, Philadelphia, Pa. The four new members of the Council were introduced as follows: J. F. Collins, Jr., Pittsburgh, E. N. McDonnell, Chicago, T. H. Urdahl, Washington, D. C., and C.-E. A. Winslow, New Haven.

President Giesecke then announced the final paper on the program The Performance of Stack Heads by D. W. Nelson, D. H. Krans and A. F. Tuthill (complete paper published in January 1940 ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*), which was given by Professor Nelson. Comments were offered by J. H. Van Alsburg.

H. H. Erickson, Philadelphia, chairman of the Committee to Study Method of Selecting Society Officers and Council Members then presented his report on procedures for the Nominating Committee, which he moved be sent to the membership for consideration before final adoption. The motion was seconded and unanimously adopted.

The report of the Resolutions Committee was then presented by A. F. Nass, Pittsburgh.

Resolutions

Whereas, the 46th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS held in the city of Cleveland has been an outstanding success, be it hereby resolved that the following resolutions of thanks and appreciation be spread on the minutes of the Society and copies thereof be transmitted to each of those persons and agencies contributing thereto.

The City of Cleveland for its hospitality.

The Honorable Harold L. Burton, Mayor of the City of Cleveland, for his cordial greeting.

The Cleveland Convention and Visitors Bureau for its cooperation and assistance.

The Committee on Arrangements of the Cleveland Chapter for its finely planned program.

The City press and trade papers for their generous coverage of our sessions.

The Management of the Exposition for the largest show of heating, ventilating and air conditioning equipment.

The authors of technical papers for their fine contributions.

The Speakers who have addressed us so entertainingly.

To the Hotel Management and staff for their excellent service, and

To all those who have contributed so much to our pleasure and enjoyment while we have been in Cleveland.

It was moved and seconded, that the resolutions be unanimously adopted.

President Giesecke called for any new business, and there being no further business, declared the 46th Annual Meeting adjourned at 11:40 a. m.

Little Men (and Women)

You've Had a Busy Week

There were many early arrivals in Cleveland for the 46th Annual Meeting of the Society, and a very heavy registration was recorded on the first day, January 22. The entire day was devoted to committee meetings, Conference of Chapter Delegates, and a meeting of the Council.

Promptly at 9:30 a.m. the Conference of Chapter Delegates convened in the Pine room and representatives of all 29 Chapters were present when President McIntire gave a brief speech of welcome. Earle W. Gray, Oklahoma City, served as chairman and S. W. Boyd, Atlanta, was secretary of the Conference.

At 10:00 a.m. a meeting of the Council was held and reports of the five Council Committees were presented and approved. Reports of special committees were received and several matters of business were referred to the incoming Council for final action. Resignations of 48 members were accepted, and 12 members were dropped from the rolls for non-payment of dues. Resolutions of appreciation to President McIntire and the retiring Council members were adopted.

At 2:00 p.m. the Officers and Council assembled at Lakeside Hall for the formal opening of the 6th International Heating and Ventilating Exposition, at which 303 exhibitors participated, and which during the week attracted the attention of over 25,000 people.

The Committee on Arrangements of Northern Ohio Chapter were on hand to greet the arrival of members and guests, lead

by chairman L. T. Avery, honorary chairman G. L. Tuve, and vice-chairman, Philip Cohen and L. S. Ries.

At 5:00 p.m. a social hour was enjoyed in the Pine room and the program was in charge of H. E. Wetzell and his committee.

After a full day of committee meetings members and guests assembled at 10:00 p. m. in the Euclid Ballroom of Hotel Statler, for a get-together party. The 550 present spent a pleasant evening meeting old friends.

The special program for ladies started on Tuesday morning with a visit to Severance Hall where a rehearsal of the Cleveland Orchestra was held. At noon the ladies enjoyed luncheon at Crosby's restaurant, and during the afternoon visited the Electrical Housekeeping Institute at Nela Park. On Tuesday evening an informal social hour for members and ladies was held in the Pine room, and at 6:30 p.m. 12 past presidents of the Society gathered for their annual dinner. During the evening many members and ladies enjoyed informal theater parties.

At 10:00 p.m. a stag smoker sponsored by the *National Warm Air Heating and Air Conditioning Association* was held at the Hollenden Hotel and several hundred enjoyed the festivities.

On Wednesday morning the ladies Committee arranged a shopping tour and in the afternoon the ladies enjoyed a bridge party at Halles Tea Room, after which a fashion show was given.

On Wednesday evening the 46th annual banquet of the Society was held in the grand ballroom of the Statler. At the conclusion of an excellent dinner, toastmaster Walter Klie introduced W. H. Driscoll, who presented the retiring president, J. F. McIntire, with the past president's emblem. Mr. McIntire responded briefly and the president-elect, F. E. Giesecke, was then introduced. Two brief, but entertaining talks were given by Dr. W. E. Wickenden, president, Case School of Applied Science, and Dr. Harvey N. Davis, president, Stevens Institute of Technology, Hoboken, N. J., whose address covered the subject of The Engineer's Job Today.

On Thursday many members and guests attended the luncheon of the Rotary Club of Cleveland, and heard Dr. C.-E. A. Winslow, vice-chairman of the Committee on Research, talk on The Air-Cooled Human Body. His address, which was broadcast, was heard by 300 Rotarians and their guests.

On Thursday afternoon there were a number of inspection trips arranged by Prof. C. A. McKeeman, chairman of the Inspection Trips Committee. Two visits were made to the Industrial Rayon Corp., Painesville, and a group of 45 saw the new plant of Republic Steel Corp. Smaller groups made a tour of inspection of the *AGA* Laboratory and the mechanical engineering department laboratory of the Case School of Applied Science.

At the conclusion of the meeting everyone who attended was lavish in his praise of the work of the Committee on Arrangements, and resolutions were adopted for the many courtesies extended by the members of the Northern Ohio Chapter.

MAIN ENTRANCE
6th INTERNATIONAL HEATING & VENTILATING EXPOSITION
6th INTERNATIONAL HEATING & VENTILATING EXPOSITION

Opening Day at Exposition

Group of ASHVE Members and Friends

Exposition Successful in Cleveland

Startling developments in air conditioning, heating, ventilating, refrigerating and associated fields were emphasized at the 6th International Heating and Ventilating Exposition held during the week of January 22 to 26, at Lakeside Hall, Cleveland. A total of 303 leading manufacturers presented comprehensive displays of the latest air conditioning equipment and accessories, making this the largest Exposition of its kind ever held. An enthusiastic highly selected audience totaling nearly 25,000 visitors came from all parts of the United States, Canada, and from many foreign countries to see this unusual review of engineering advance. Unusual buying interest was reported by many exhibitors, some of whom made actual sales of display equipment during the exposition.

The Exposition, otherwise known as the Air Conditioning Exposition, was held under the auspices of the American Society of Heating and Ventilating Engineers, who, following an established custom, held its 46th Annual Meeting during exposition week. Meeting also in Cleveland was the *National Warm Air Heating and Air Conditioning Association.*

The first heating and ventilating exposition was held in Philadelphia in 1930 and subsequent presentations have been in Cleveland, Chicago, and New York. More than 70 per cent of the space at this year's Exposition was taken by firms who previously had participated.

NWAHACA Elects Officers in Cleveland

L. R. Taylor, president, *National Warm Air Heating and Air Conditioning Association,* in welcoming the members and friends to the 44th convention held at the Hotel Hollenden, Cleveland, January 22 to 24, stressed the advancement made by the association and the importance of the work done by it. Allan P. Ames discussed the amount and value of publicity received by the association and its work. Other talks given at the opening session were by Charles M. White, Cleveland, C. Ackerson, Gadsden, Ala., and C. T. Burg, Cleveland.

The officers elected for the coming year are as follows: president, C. A. Olsen, Elyria, Ohio; 1st vice-president, H. S. Sharp, Cleveland; 2nd vice-president, Shirley Percival, Des Moines, Ia.; and Allen W. Williams, Columbus, was re-elected managing director and treasurer.

In the first of the research sessions, F. G. Sedgwick, Minneapolis, chairman of the association's research advisory committee, reported on the work of the committee, outlining current research, and sketching contemplated research activity. Two technical papers given at this session were Heat Insulation, presented by A. P. Kratz, and Resistance of Air Flow to Regis-

ters and Grilles, delivered by S. Konzo, both from the University of Illinois.

B. F. McLouth, Lansing, Mich., chairman of the installation codes committee, outlined the association's codes, telling of their value and in general pointed out the progress that had been made in the work of the committee. J. H. Van Alsburg, Chicago, presented a paper on Sizing Registers for Forced Air and Air Conditioning Installations, emphasizing the importance of correct sizing and location of registers for proper performance of either a forced air or air conditioning system. J. S. Irvine, Toledo, spoke on the over-all merchandising interest of the association. An outline of the 9th annual Short Course to be held March 25, at Michigan State College, was given by Prof. L. G. Miller, of the mechanical engineering department of the College. He extended an invitation to attend the course and told of its advantages to all those interested in the sale and installation of forced air and air conditioning.

The final session was held jointly with the ASHVE, in connection with their 46th Annual Meeting also convened in Cleveland at this time.

A Distinguished Member

On the Society's rolls there are 84 members who were born in the year the Society was organized 1895, but one member has the unique distinction of being born on the day of the Society's incorporation, January 24, 1895, and this member is Joseph Davis, Buffalo, N. Y.

John James Addresses Western Chapters

Speaking on What Research Means to Better Comfort, Efficiency and Health, John James, technical secretary of the Society, visited four western chapters recently, including Oregon, Pacific Northwest, Manitoba, and Minnesota.

In his talk, Mr. James emphasized how the joint efforts of the physician, bacteriologist, physiologist, psychologist and engineer are being coordinated through available scientific research channels to determine the standards for better living. He also stated that heat, humidity and air movement have long been established as important criteria of comfort, but today such additional factors as noise, color and radiation are felt to have some significance in appraising optimum environmental conditions. He further mentioned that through investigations on the removal of atmospheric impurities and air-borne bacteria in the treatment of indoor air, it is expected that improved health standards will result and a great deal of human suffering will be alleviated.

Death of Charles S. Sonneborn

Word has just been received of the death of Charles S. Sonneborn, New Castle, Pa., on November 4, 1939. Mr. Sonneborn was born in Hannover, Germany, in 1874, and came to this country in 1891, locating in Philadelphia, Pa. Later he moved to Pittsburgh and was employed by the Union Switch and Signal, and Westinghouse Electric Mfg. Co., Swissvale, Pa.

His inventive genius started him in his own business with the Pittsburgh Instrument and Machine Co., where he worked not only on his own inventions, but did considerable model work for others.

In 1904 he was called to the Garland Mfg. Co., as superintendent of their nut and bolt plant, and in 1914 he was placed in full charge of their pressed metal radiator division.

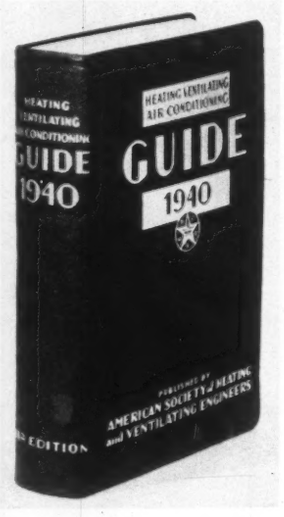
In 1926 he became vice-president in charge of production and design of Shaw-Perkins Mfg. Co., West Pittsburgh, Pa. Mr. Sonneborn joined the ASHVE in 1930, and was always fully appreciative of the value of the research carried on by the Society.

The Officers and Council of the Society extend their sincere and heartfelt sympathy to his widow, who survives. Mr. Sonneborn was active in Society affairs and will be missed by his many friends and associates among the members.

The Guide 1940 Released to Members

Heating, Ventilating, Air Conditioning Guide 1940, size 6x9 inches, and 1184 pages, is now being distributed to members of the Society.

Prepared under the direction of the Guide Publication Committee, it is an authoritative reference manual for the profession and industry, now in its 18th edition, issued annually and devoted exclusively to the subjects of heating, ventilating and air conditioning, and related data on refrigeration. Compiled by men active in the profession and the allied industries, it is arranged in two sections—a Technical Data Section of 46 chapters and a Catalog Data Section on manufacturers' equipment.



It supplies, in one compact volume, engineering data and information on modern equipment, all of which is essential for engineers, architects, contractors and others engaged in the industry.

This new volume is less bulky and lighter in weight than previous editions, due to condensation and consolidation of text matter, and the use of lighter paper. Although there has been addition of much new data in old chapters and one new chapter, the result is a volume actually containing more information, but easier to handle and more convenient to use. A chapter index and a detailed subject index give ready reference to the technical data.

The Catalog Data Section has been entirely re-arranged. The many products described by manufacturers have been grouped into five divisions and the contents of each division outlined on a title page, with page references to each group of products, and cross references to relevant matter in the Technical Data Section. An Index to Modern Equipment provides a convenient classified reference to manufacturers' data on apparatus and materials.

The Guide Publication Committee, F. C. McIntosh, *Chairman;* P. D. Close, C. M. Humphreys, A. J. Offner, and S. S. Sanford, are to be congratulated for this fine volume.

Pittsburgh Members Inspect Buhl Planetarium

February 12, 1940. The February meeting of the Pittsburgh Chapter was devoted to an inspection trip through the Buhl Planetarium and Institute of Popular Science. Following dinner at the Roosevelt Hotel, the party inspected the air conditioning system of the Planetarium. The system was described briefly by the designing engineer and then Mr. Schumacher, the building superintendent, and Mr. Bensen, Johnson Service Co., acted as guides and conducted the groups through the machinery rooms.

Most of the party then attended the regular evening performance of the "Sky Show", after which the members of the Society and their guests were invited to remain for a demonstration of the operation of the Planetarium Projector, an event

which proved to be the most enjoyable part of the evening's program.

There were 75 members and guests in attendance at this meeting as reported by T. F. Rockwell, secretary of the Chapter.

Open Forum on Insulation Held at Ontario

February 5, 1940. Fifty-one members and guests of the Ontario Chapter of the Society gathered at the Royal York Hotel, Toronto, for their February meeting, at which Pres. H. D. Henion presided.

Following the reading of the minutes of the preceding meeting which were accepted as read, President Henion called upon H. H. Angus to preside at an open forum on insulation. Mr. Murray, of the Building Products Ltd., spoke on fiber board insulation, followed by Mr. Graham of Canadian Lime and Alabastine Co., who spoke on mineral wool insulation. E. B. Sheffield, Armstrong Cork Co., then addressed the meeting on cork board insulation.

T. Buckley, Canadian Gypsum Co., spoke briefly on reflective type insulation, while Mr. Smith, Cooksville Co., followed with the other phases of insulation.

Considerable discussion followed on the application of insulation to buildings, and before adjournment, H. G. Hill extended a vote of thanks on behalf of the Chapter to the five speakers.

Western New York Hears W. H. Driscoll

January 9, 1940. The January meeting of the Western New York Chapter was held at the University Club, Buffalo, with 40 members and guests attending.

The minutes of the two previous meetings were read and approved. Pres. L. P. Saunders congratulated Messrs. Davis, Mahoney and Strause for the successful Christmas Party, and announced that arrangements were made to have J. R. Vernon, advertising manager, Johnson Service Co., Chicago, speak at the February meeting.

B. C. Candee gave the treasurer's report, which was followed by an announcement by Roswell Farnham that preparations for the chapter dinner dance scheduled for February 22 had been completed, and it was voted to invite all members of the air conditioning council.

C. W. Farrar urged a large attendance at the Annual Meeting in Cleveland and mentioned that an effort was being made to secure a special coach for convenience to take the members to Cleveland.

L. A. Harding reminded the members that the new Buffalo Civic Auditorium would be completed in time for the 1942 Air Conditioning Show.

President Saunders introduced the guest speaker of the evening, W. H. Driscoll, past-president of the Society, who addressed the meeting on the subject of amputating the square foot, which referred to the Society's recommendations for use of the MB and MBH unit instead of the antiquated square foot basis of rating heating jobs. Mr. Driscoll also reminisced about the early days of the heating business and the origin of the ASHVE, and traced the early history of the research work and THE GUIDE, all of which the members found very interesting.

Prof. Wilkes Discusses Building Insulation at Montreal

January 15, 1940. There were 35 members and guests in attendance for the regular monthly meeting of the Montreal Chapter which was held at the M.A.A.A.

The meeting was called to order by Pres. C. W. Johnson, and T. H. Worthington extended a welcome to the guests present.

G. E. Perras moved that the reading of the minutes of the previous meeting be waived, and the motion was seconded by R. F. Wormley.

W. U. Hughes introduced the speaker of the evening, Prof. G. B. Wilkes, professor of heat engineering at Massachusetts Institute of Technology, Cambridge, Mass., who addressed the meeting on the subject of determining the effectiveness of building insulation in theory and practice. A rising vote of thanks was given to Professor Wilkes, and the meeting adjourned at 9:07 p. m.

Detroit Edison Hosts to Michigan Chapter

January 15, 1940. The Michigan Chapter was the guest of the Detroit Edison Co. for its January meeting. The meeting was held in the auditorium in the general offices of the company and 115 members and guests were present for the dinner and meeting.

Immediately following dinner the meeting was called to order by Pres. W. C. Randall. The reports of the secretary, treasurer, membership and program chairmen were read and approved. Vice-Pres. G. H. Tuttle announced that the dinner dance would be held at Dearborn Inn on April 27.

J. F. McIntire, president of the Society, spoke on the Cleveland Annual Meeting and the Exposition.

A resolution was presented expressing sincere regret on the death of Bruce Wigle, copy of which will be sent to his widow.

President Randall then introduced J. H. Walker, engineering assistant to the general manager of Detroit Edison Co., who presented the speaker of the evening, Sabin Crocker, engineer for the company, who spoke on the standards and codes for piping as applied to welded construction.

The meeting was concluded with a feature movie entitled "Mr. Topper Takes a Trip."

Western Michigan Chapter Meets

January 8, 1940. The Western Michigan Chapter held its January meeting at the Hotel Rowe, Grand Rapids, with 53 members and guests in attendance.

Pres. B. F. McLouth called the meeting to order and after the secretary read the minutes of the December meeting, reports of the various standing committees were made.

President McLouth introduced Herbert Kunen, New York, who gave an interesting illustrative talk on applications and adaptability of equipment in air distribution design.

Following considerable discussion the meeting adjourned at 10:30 p. m.

Iowa-Nebraska Members Inspect New Bankers Life Bldg.

January 9, 1940. The January meeting of the Iowa-Nebraska Chapter was held at the Chamberlain Hotel, Des Moines, Ia. After dinner was served, the meeting was called to order by Pres. T. R. Johnson, with 43 members and guests in attendance.

The minutes of the preceding meeting were read and approved, and E. H. Danielson, chairman of the membership committee, gave a very enthusiastic pep talk urging members to join the local chapter.

A. R. Brown, Leopold Engineering Co., and M. T. Johnson, business manager of the new Bankers Life Building, were introduced by A. L. Walters, who then introduced W. E. White, Pennsylvania Engineering Co. Mr. White gave a brief talk describing the design, construction and operation of the mechanical equipment in the new Bankers Life Building, and stated that from the standpoint of mechanical equipment, this building is one of the most outstanding buildings constructed in the United States in the last three or four years.

The meeting adjourned at 8:00 p. m. and reconvened at 8:10 in the lobby of the new Bankers Life Building. The group was

divided into two sections and tours were conducted by Mr. Johnson and Mr. White.

The construction and operation of the mechanical equipment in the building was explained in detail, and included radiant heating and cooling, duct supply and exhaust systems.

Most of the equipment in the building was still uncovered, presenting a very good opportunity to see and have explained the construction and operation of the most advanced installation of heating and ventilating and air conditioning equipment in the State of Iowa.

The meeting adjourned at 10:00 p. m.

Regulations of National Fire Prevention Association Discussed at St. Louis

January 9, 1940. The regular monthly meeting of the St. Louis Chapter was held at the Kingsway Hotel, with 29 members and guests present.

J. H. Carter, chairman of the attendance committee, made a brief report on the activities of his committee, and C. F. Boester read the minutes of the previous meeting, which were approved as read.

Secretary Boester then brought up the question of the new code which is being prepared by E. P. Bradley.

A letter was read from the St. Louis Convention Bureau requesting that the St. Louis Chapter join the Convention Bureau in a plea to secure the 1942 Annual Meeting of the Society for St. Louis. The consensus of opinion was that the matter should be discussed further before any final action is taken.

Pres. R. J. Tenkonohy was chosen as the official delegate of the St. Louis Chapter to the Society's Annual Meeting in Cleveland.

R. C. Smith, York Ice Machinery Corp., was called upon to discuss the paper on air metering devices presented at the Fall Meeting of the Society, Atlanta, which was followed by an interesting discussion.

Edward Sestak, Missouri Inspection Bureau of *National Board of Fire Underwriters,* presented a paper on the subject, "A Discussion of Regulations of the National Fire Prevention Association Governing the Installation of Air Conditioning, Warm Air Heating, Air Cooling and Ventilating Systems Employing Ducts." Mr. Sestak's talk aroused a spirited discussion, and he was requested to send the secretary several copies of the paper.

The meeting adjourned at 10:00 p. m.

Air Distribution Design Topic at Atlanta

January 3, 1940. The first regular meeting of the 1940 season was held at the Cox-Carlton Hotel, with 18 in attendance.

Pres. T. T. Tucker called the meeting to order and a discussion followed on the revised by-laws as recommended by the by-laws committee and submitted to the membership for consideration. A motion was made and unanimously passed to adopt the revised by-laws, changing Article 2, Section 5 to require 8 members for a quorum rather than 5.

The chairman of the research committee reported that he had written to Prof. F. E. Giesecke, A. & M. College of Texas, College Station, for his advice with reference to problems for research at Georgia School of Technology.

Dr. R. L. Sweigert of Georgia-Tech. announced a citizenship institute scheduled in Atlanta for February 4 to 6, and impressed upon all present the necessity for greater thought and action by engineers in the duties of citizenship.

Following the report given by the treasurer, the speaker of the evening, Herbert Kunen, New York, gave an interesting talk on air distribution design, which was followed by a series of questions which Mr. Kunen answered. The meeting was adjourned following the discussion.

Fred R. Still

Dies of Pneumonia

On February 15, at the age of 73 Fred R. Still, vice-president of the American Blower Corp., in charge of exports, died of pneumonia at his home 3457-82nd St., Jackson Heights, L. I. He was born at Saginaw, Mich., November 25, 1867 and received his early education in ordinary home schools. He started work at the age of 17 as a draftsman, and in 1887 became associated with Huyett & Smith Mfg. Co., the predecessor of the American Blower Corp. He served as chief draftsman and in almost every capacity with the company, in the shop, in the field and in the office during a career of 53 years of continuous service. In 1895 he became chief engineer and directed the work of the company in the manufacture of heating, ventilating and drying apparatus, fans, blowers, and engines etc.

In 1904 he joined the Society and took a very active part in its work. As his membership spanned all but 10 years of the Society's existence, he witnessed the tremendous development of the profession and industry and the growth of the Society from a total membership of 220 to nearly 3200. His was an active role in the work of the Society, and its TRANSACTIONS reveal his contributions in papers which he delivered before the Society, and the service that he rendered as a member of numerous Committees on the Council and as an officer. He was instrumental in the founding of the Michigan Chapter in 1916, and always took an active part in its affairs.

At the Society's 23rd Annual Meeting in New York, 1917, Mr. Still took part in the preliminary discussion which resulted in the appointment of the committee to investigate the possibility of establishing a research laboratory. This committee reported its recommendations at the Semi-Annual Meeting in Chicago, and during 1918 when Mr. Still was president of the Society, the committee made its final report which resulted in the establishment of the Society's Research Laboratory at Pittsburgh, in 1919 with the late Prof. John R. Allen, dean of engineering at the University of Minnesota, as director.

Mr. Still served as chairman of numerous committees in the Society; in 1917 on the Committee on Charter for New Chapters; in 1919 as chairman of the Committee on Research; and as chairman of the Executive Committee; in 1920 as chairman of the Nominating Committee and again as chairman of the Committee on Research; in 1922 as chairman of the Sub-Committee on Subjects (Committee on Research); and as vice-chairman of Committee on Standards of Ventilation; he served as President of the Society and chairman of the Council in 1918. He was instrumental in establishing THE GUIDE in 1922 and contributed to its text section on many occasions.

Mr. Still was recognized as one of the country's leading authorities on heating, ventilating and air conditioning. Always ardently interested in scientific research, he developed and perfected new types of machinery and processes in those fields of engineering.

As a youth he was one of the expert sailors on Saginaw Bay. This activity continued after he moved to Detroit, where he participated in racing competition of both sailing and power boats and gained particular recognition as a navigator. He was Commodore of the Detroit Motor Boat Club and became an outstanding racing official in the power boat field. He was chairman of the regatta committees of the Gold Cup races held in Detroit and of the *Inter Lake Yachting Association* for several years, and in charge of the Perry Centennial Regatta. He served as

president of the *American Power Boat Association* from 1923 to 1931.

In addition to his membership in the ASHVE and the *American Power Boat Association*, Mr. Still was a member of the *American Society of Mechanical Engineers* and the *American Society of Naval Architects and Marine Engineers.*

The Officers and Council of the Society are saddened by the loss of an old and loyal member, and extend their sincerest sympathy to his widow, Mrs. Harriet Humphrey Still, and son, Roscoe Wilson Still, who survive.

Death of James W. Partlan in Detroit

James W. Partlan, president of the heating and ventilating company bearing his name, died on February 2, 1940, at his home 2521 Edison Avenue, Detroit, after a brief illness. Mr. Partlan was 81 years old and had been a lifelong resident of the city of Detroit. He received his early education in the ordinary home schools, and his apprenticeship with the Detroit Metal and Heating Works, Detroit. In 1889 he went into business for himself as contracting engineer in heating, ventilating and plumbing.

Mr. Partlan joined the ASHVE in 1916, and at the Annual Meeting of the Society in Cleveland, 1932, it was by a unanimous vote of the Council that he was designated a Life Member of the Society.

He is survived by his widow, Mrs. Sallie Partlan, a son, James S., a partner in the company, and a brother, C. M. Partlan, to whom the Officers and Council extend their sincere sympathy.

Mr. Partlan, besides holding membership in the Society, was a member of the Detroit Athletic Club, the Detroit Golf Club and the Detroit Curling Club.

Kansas City Member Dies in Florida

Benjamin Natkin, president of Natkin & Co., Kansas City, Mo., died suddenly at Miami Beach on January 1, 1940, where he had gone to regain his health.

Mr. Natkin came to the United States as a boy, and following his graduation in 1905 from the Armour Institute of Technology, Chicago, he began his work in the heating specialty field in which he continued until his death. He came to Kansas City in 1907 and founded the firm of Natkin Engineering Co., which, under his guidance, enjoyed an enviable growth having offices extending from Nebraska through the state of Texas. At the time of his death he was active head of the firm which had offices in Kansas City, St. Louis, Omaha and Tulsa, Okla.

He was one of the pioneers in the engineering, design and sale of the vapor vacuum heating system, and in the heating and air conditioning industry he gained a national reputation.

Among Mr. Natkin's varied activities was the promotion and construction of the President Hotel, one of Kansas City's leading hostelries.

Mr. Natkin joined the ASHVE in July 1907, and was among those who signed the petition for the Kansas City Chapter in 1916. He became president of that Chapter in 1922, and remained active in Society affairs until his death.

He is survived by his widow, Mrs. Selma M. Natkin and two sons, Bertram Natkin, St. Louis, and Alfred Natkin, Tulsa, Okla., to whom the Officers and Council extend their heartfelt sympathy.

Edward Moss Dies

Edward Moss, a member of the ASHVE since 1919, died at his home, Hollis, L. I., on February 5, 1940 after a prolonged illness. He was engaged in the heating and ventilating business in Brooklyn for over fifty years, from 1886 to 1890 as an apprentice with James Armstrong Co., then after working two years as a journeyman steam fitter, he started in business for himself as a plumbing and heating contractor. In 1896 he entered the employ of the Brooklyn Rapid Transit Co., as superintendent of the plumbing and heating department, where he designed and planned, and supervised the installation of the heating and ventilating systems installed in their various depots, office buildings, car barns and stations. After more than 25 years with the B. R. T. he became associated with the New York Rapid Transit Corp. where he remained until his death.

Mr. Moss took a great interest in the Society activities and will be missed by his many friends and associates in the ASHVE. The Officers and Council extend their sincere sympathy to his widow and brother, who survive.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 37 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge the members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by March 15, 1940, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
ARMOUR, E. G., Sales Engr., Armour's, Brantford, Ont., Can. (*Advancement*)	J. P. Fitzsimons C. M. Humphreys	T. F. Rockwell J. F. Collins, Jr.
BERTOLETTE, CHESTER, JR., Htg. & Air Cond., Modern Automatic Heat Co., Poughkeepsie, N. Y.	G. C. Burr F. H. Oertel	W. M. Wallace, II Harry Hoffman

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
BOWES, W. H., Sales Engr., Canadian General Elec. Co., Ltd., Toronto, Ont., Can.	H. R. Roth	E. J. Davis
BROWN, L. S., JR., Student, Catholic Univ. of America, Washington, D. C.	H. G. Hill	J. P. Fitzsimons
CARTER, A. W., Htg. Engr., Monarch Brass Mfg. Co., Toronto, Ont., Can. (*Advancement*)	W. K. Karsunky	F. M. Thuney
COLE, C. B., Contracting Engr., Boynton Cole & Co., Atlanta, Ga. (*Advancement*)	M. T. Firestone	F. A. Leser
CRARY, J. O., Mgr. Air Cond. & Commercial Dept., Frigidaire Div., General Motors Sales Corp., New Orleans, La.	H. B. Jenney	F. E. Ellis
CROLEY, J. G., Htg. Engr., Savannah Gas Co., Savannah, Ga.	William Philip	S. A. Jennings
	C. L. Templin	T. T. Tucker
GEBEL, K. M., Student, C. C. N. Y. School of Tech., New York, N. Y.	L. F. Kent	W. J. McKinney
GOLDMANN, PHILIPP, Student, Catholic Univ. of America, Washington, D. C.	G. E. May	F. S. McLaren
GRANT, W. H., JR., Br. Mgr., Warren Webster & Co., New Orleans, La.	D. J. Purinton	F. G. Burns
GRAVES, F. V., Engr. & Sales, Frank A. Leon Co., Washington, D. C.	J. B. Hamlin, Jr.	H. C. Mills
GRAY, W. C., Air Cond. & Htg. Engr., Gulf, Mobile & Northern Railroad Co., Bogalusa, La.	J. E. Swenson	R. T. Haley
GUILBERT, S. R., Air Cond. Sales Engr., The Riester & Thesmacher Co., Cleveland, Ohio	F. W. Hanburger	H. F. Frese (*Non-Member*)
HACKETT, F. C., Sales Repr., Bell & Gossett Co., Washington, D. C.	F. L. Hilder	L. F. Goltz (*Non-Member*)
HEAGERTY, W. H., Sales & Engrg., Wm. H. Heagerty Co., Washington, D. C. (*Reinstatement*)	W. K. Karsunky	F. M. Thuney
HESSELSCHWERDT, A. L., Jr., Instructor, Wayne Univ., Detroit, Mich. (*Advancement*)	M. F. Firestone	F. A. Leser
JENKINSON, V. J., Sales Engr., Minneapolis-Honeywell Reg. Co., Ltd., Toronto, Ont., Can.	L. V. Cressy	J. J. Friedler, Jr.
JUERGENS, W. A., Sales Engr., John P. Childe Co., Cincinnati, Ohio	G. E. May	N. J. Helwick
KING, R. W., Sales Engr., Asbestos Covering & Roofing Co., Washington, D. C.	William Bornstein	R. H. Feltwell
LUDLOW, H. M., Sales & Engrg., Jackson, Miss.	W. E. Kingswell	J. E. Axeman
	G. W. F. Myers	A. W. Clarke (*ASME*)
McCUSKER, J. P., Student, Catholic Univ. of America, Washington, D. C.	W. J. Oonk	H. C. Morrison (*ASRE*)
McENTEE, F. M., Air Cond. Engr., Archt. of the Capitol, Washington, D. C.	L. E. Slawson	H. E. Wetzell
MEHMKEN, H. O., Engr., York Ice Machinery Corp., St. Louis, Mo.	C. M. H. Kaercher	C. A. McKeeman
MOFFAT, O. G., Application Engr., Canadian Westinghouse Co., Ltd., Hamilton, Ont., Can. (*Advancement*)	D. B. Tuxhorn	W. H. Littleford
MOLLANDER, E. D., Dir., Register & Grille Mfg. Co., Inc., Brooklyn, N. Y.	W. E. Kingswell	W. D. Marshall
MORRO, J. J., Engr. in Charge, Paragon Oil Burner Corp., New York, N. Y.	W. H. Littleford	F. M. Thuney
NORTON, J. A., Mgr. Htg. Section, Crane Ltd., Toronto, Ont., Can.	E. J. Febrey	F. A. Leser
OWINGS, H. L., Refrig. & Air Cond. Engr., Houston Natural Gas Co., Houston, Tex.	J. H. Walker	James Holt
PAETZ, G. A., Student, Purdue Univ., West Lafayette, Ind.	J. F. McIntire	D. L. Perkins (*ASME*)
	H. G. Hill	Ernest Fox
PAULEY, R. D., Development Engr., Wood Conversion Co., Cloquet, Minn.	H. D. Henion	J. P. Fitzsimons
REPP, H. L., Br. Mgr., U. S. Radiator Corp., Cleveland, Ohio (*Reinstatement*)	R. P. Whitmer	J. E. Allen (*Non-Member*)
RESCH, R. J., Pres., McQuay, Inc., Minneapolis, Minn.	H. A. Soper	W. D. Ehlers (*Non-Member*)
	E. J. Febrey	W. D. Marshall
STANGLE, W. H., Special Engr., Servel Inc., New York, N. Y.	W. H. Littleford	T. L. Eagan (*Non-Member*)
	M. F. May	R. E. Pauling
STAUFFER, J. E., Student, Carnegie Inst. of Tech., Pittsburgh, Pa.	R. C. Malvin	H. L. Mohn
TILLINGHAST, H. S., Estimator, Minneapolis Gas Light Co., Minneapolis, Minn.	W. K. Karsunky	F. M. Thuney
WYNGARDEN, J. E., Mgr., Day Heating Co., Salem, Ore.	M. T. Firestone	F. A. Leser
	G. D. Fife	J. J. Nolan, Jr.
	M. F. Hoppe	S. P. Eagleton
	L. J. Du Bois	J. M. Musser (*Non-Member*)
	A. G. Matousek	H. E. Miller (*ASRE*)
	James O'Neil	J. P. Fitzsimons
	H. R. Roth	A. S. Leitch
	A. C. Buensod	H. L. Janet
	J. C. Rose	C. S. Pabst
	B. H. Schulze	W. M. Heebner
	L. W. Charlet	V. J. Cucci
	H. R. Roth	G. E. Cole
	M. W. Shears	S. A. Jennings
	I. E. Rowe	D. S. Cooper
	C. A. McKinney	B. P. Fisher
	W. T. Miller	A. W. Cole (*ASME*)
	W. J. Cope (*ASME*)	L. V. Ludy (*ASME*)
	R. E. Backstrom	A. B. Algren
	D. B. Anderson	H. M. Betts
	J. E. Wilhelm	C. M. H. Kaercher
	L. E. Slawson	E. W. Gray
	E. H. Seelert	C. L. Bensen
	J. F. Stafford	M. L. Benson
	A. J. Offner	C. H. B. Hotchkiss
	R. H. Carpenter	W. E. Heibel
	C. M. Humphreys	G. G. Waters
	T. F. Rockwell	F. C. McIntosh
	J. E. Swenson	R. C. Jordan
	F. B. Rowley	C. E. Lund
	B. W. Moore	J. D. Kroeker
	W. R. Barton	B. W. Farnes

CANDIDATES ELECTED

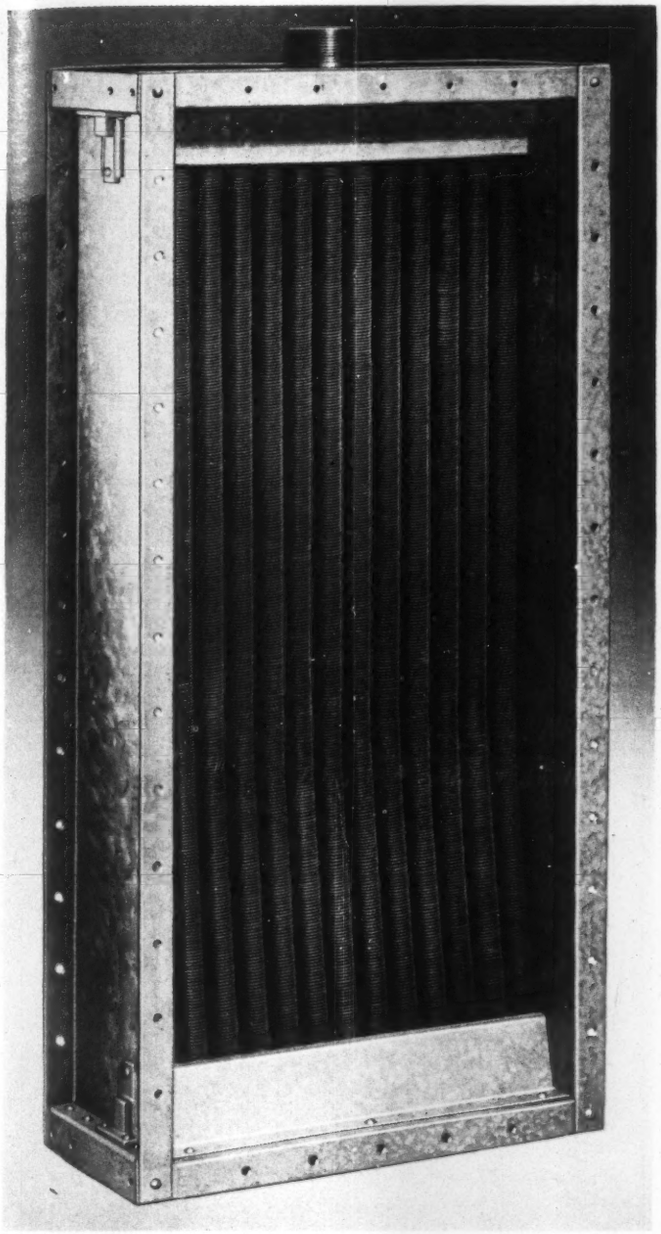
In the past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. B-III, Sec. 8, of the By-Laws, the following list of candidates elected:

MEMBERS

COTT, W. B., Air Cond. Engr., The Doermann-Roehrer Co., Cincinnati, Ohio.

ASSOCIATES

FUNK, D. S., Pres., National Standard Air Products Co., Chicago, Ill



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☆ ☆ ☆ Heating men will find that the design of Fedders Type K Heating Coils incorporate the newest, most up-to-date and important improvements offered to their industry. They are designed by engineers who recognize and appreciate the balance between practical and theoretical design and their influence on heat transfer and air handling efficiency as well as long life under severe operating conditions.

Type K Heating Coil features include patented full-floating relief of over-all expansion stresses, and "knee action" relief of differential expansion between individual tubes . . . spiral fins are pressure wound and metallically bonded on seamless copper tubes . . . entire surface is tinned for protection against corrosion . . . integral orifices and scale breakers . . . generously large cylindrical headers.

Husky, rigid ARMCO ZINCGRIP casings assure adequate protection against piping stresses and coil distortion. Casings are only 9″ deep,—made possible by Fedders exclusive methods of full expansion relief. They provide compact size, extreme strength, and long life, trouble-free service.

CATALOGED in an Easy to use Data Book!

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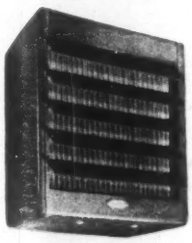
CONCERN .....

STREET .....CITY.....



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Fedders handsome, sturdy cabinets . . . quiet operation . . . exclusive streamline tubes and non-clogging fins are approved by Heating Experts everywhere. Built in a complete line from 75 to 1100 EDR.

Dependable Service

You'll have plenty of reasons to be grateful if you install Fairbanks Iron Body Valves. They are sure to give years of trouble-free service, for their quality is deep down and enduring.

The easy manipulation . . . ease of repacking . . . ease of renewing . . . interchangeable parts . . . sturdy construction . . . perfect seating . . . expert workmanship . . . careful testing . . . these are but a few of the features that will appeal to you.

For more than 50 years it has been our constant aim to make valves that satisfy from the standpoint of operation, durability and economy.

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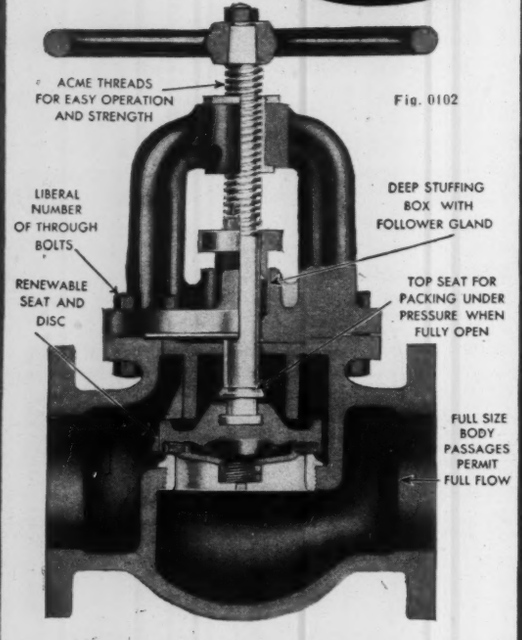

ACME THREADS FOR EASY OPERATION AND STRENGTH
Fig. 0102
LIBERAL NUMBER OF THROUGH BOLTS
DEEP STUFFING BOX WITH FOLLOWER GLAND
RENEWABLE SEAT AND DISC
TOP SEAT FOR PACKING UNDER PRESSURE WHEN FULLY OPEN
FULL SIZE BODY PASSAGES PERMIT FULL FLOW

Equipment Developments

For your convenience in obtaining more information about any of this equipment, see coupon on page 150. Add the new products and companies listed here to your Directory Section which you received in your January, 1940, HEATING, PIPING AND AIR CONDITIONING and thus keep your records of sources of supply up to date throughout the year . . . Single asterisk (*) indicates equipment not listed in Directory Section; double asterisk (**) equipment and manufacturer not listed

Vertical Delivery Unit Heaters

No. 1787—Eleven models of a new line of vertical delivery unit heaters are available in a capacity range from 30,000 Btu per hr (125 EDR) to 471,600 Btu (1965 EDR). These heaters are recommended by the maker for applications where conditions call for a directly downward delivery of heated air; in factories where high clearance is essential, as for craneways, over store and office doorways, etc.

Each unit is regularly equipped with a radial or spoke type deflector assembly known as the "Cone-Jet Deflector." By manual adjustment of the individual deflector blades, the delivered air may be concentrated in limited direction or kept almost entirely to one side of the unit. Thus, in a multiple installation heated air deflection may be controlled so as to give more concentration on outside walls. By employing a steep angle of pitch on the deflectors, a high, narrow, jet like cone of heated air may be thrown from a high elevation, according to the maker. When the unit is mounted relatively close to the floor, deflectors can be flattened for delivery of a low, broad cone of heated air. A square condenser element is used. The units may be suspended directly from the supply branch.

The deflector assembly is mounted on a plate, the "Inhalator," which is separated by a 1 in. space from the bottom of the unit proper. This device "inhales" extra air in order to give greater volume and greater velocity of air movement.—Modine Mfg. Co., 17th St., Racine, Wis.

Turbo-Compressors for Air Conditioning

No. 1788*—First public announcement of a new turbo-compressor for air conditioning and refrigeration was made last month. The manufacturer has long recognized the need for supplementing its line of vertical reciprocating compressors on steam drive installations. Standard systems are available ranging in capacity from 150 to 1000 tons of refrigeration.

The new compressor is particularly suitable for steam drive, but can be powered by electricity. Its compactness is valuable in large human comfort or industrial air conditioning systems or in water or brine cooling installations for manufacturing plants, the maker states. Trichloromonofluoromethane ("Freon-11") is the refrigerant. The assembly may be mounted on upper floors, or light simple foundations.

Features include impellers operating at low speed, "balance disc" compensation of impeller thrust, and bladeless diffuser passages for reducing pumping limit and increasing efficiency. Condensers are of non-ferrous internal construction. The water cooler is of the flash type, non-ferrous construction, with hermetically sealed liquid pumps.

Control systems vary with the type of prime mover employed. Direct drive, steam turbine powered compressors are equipped with governors for speed regulating. Secondary control of variable speed motors, driving through speed increasing gears, permits flexibility when electricity is employed. Safety devices are provided to assure constant flow of chilled water and to prevent freezing.

These compressors were developed through the combined efforts of the engineering organizations of York and Allis-Chalmers Mfg. Co. Installations include the United Gas and Pipe

Advantages OF NEW DETROIT DURA-FRAM EXPANSION VALVES

They combine all the advantages of gas charged power elements with those of single diaphragm construction.

They may be located wherever convenient, even in a refrigerated compartment, without sacrificing the advantages of gas charging.

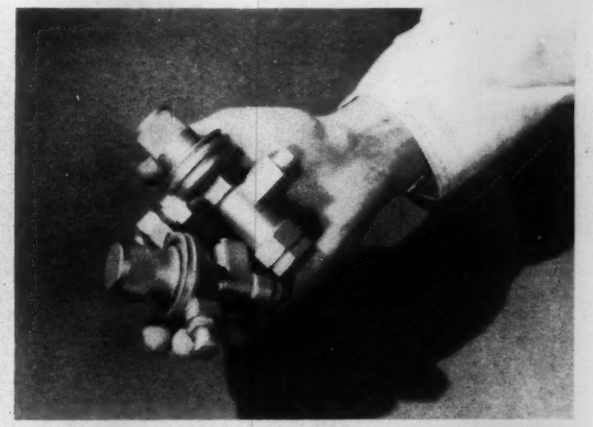
They have silver soldered joint between body casting and power element dome to prevent any leakage.

They are smaller and easier to install.

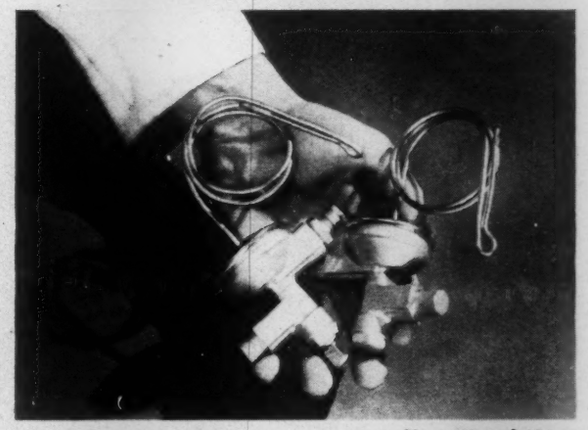
In the larger sizes they pre-cool high side liquid before reaching the orifice, thereby reducing the volume of any entrained gas, passing a more solid stream through the orifice and improving balance in the system.

The larger valves may have their capacities changed on the job merely by changing the selective capacitators or the seat assembly.

Write for detailed information.

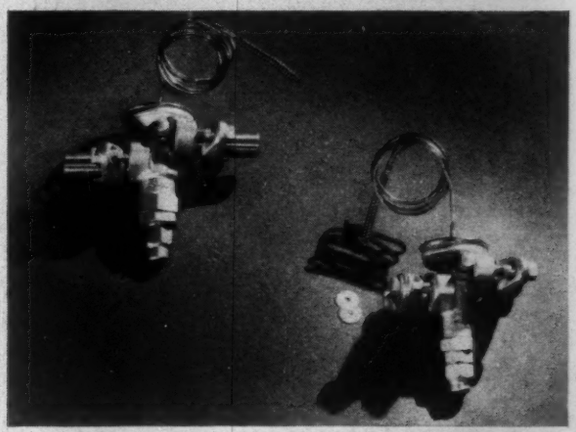


Nos. 892 and 895



Nos. 893 and 894

Nos. 896 and 897



Nos. 787 and 788

DETROIT LUBRICATOR COMPANY

General Offices: DETROIT, MICHIGAN

Canadian Representatives—RAILWAY AND ENGINEERING SPECIALTIES LIMITED, Montreal, Toronto, Winnipeg

AIR Conditioned BY VILTER

IN CLEVELAND ITS THE HOLLENDEN
IN COLUMBUS ITS THE NEIL HOUSE
IN AKRON ITS THE MAYFLOWER
IN JAMESTOWN, N.Y. ITS THE JAMESTOWN AND
THE SAMUELS
IN CORNING, N.Y. ITS THE BARON STEUBEN

Office of
THEO DEWITT
HOTEL HOLLENDEN
CLEVELAND

November 2, 1939

Vilter Mfg. Co.,
Swetland Bldg.,
Cleveland, Ohio.

Gentlemen: Attention: Mr. H.D.Andress

Now that we have had an opportunity to use the air conditioning equipment you installed at the Neil House for over a period of ninety days, I feel that I am in a pretty good position to report to you about it.

No doubt, you will be pleased to learn that this equipment functioned 100% from the minute your engineers turned same into service until just a few days ago when our engineers shut off the compressors because we do not need any air cooling at this time of the year.

We think the performance of this equipment is remarkable, compared with some experiences we have had in the past with other equipment.

I wish to take this opportunity to thank you for the interest you took in this job, as well as to assure you I will always call on you first whenever I am in need of further air cooling equipment.

Very truly yours,

TDeW:m

"Remarkable Performance"

Says Mr. Theo De Witt
Operator of Famous Hotels

★One Vilter Installation Deserves Another . . . . .

When Mr. De Witt installed "Air Conditioning—by Vilter" in his Hotel Hollenden at Cleveland, it was called the "Sweetest Job In The Country" by engineers.

That's one good reason why Vilter received the call to Air Condition at the famous "NEIL HOUSE" in Columbus, Ohio, another Dewitt-operated Hotel. And it's this same "100% Performance" that inspires Vilter Air Conditioning wherever Human Comfort, carefully and economically controlled, will pay big dividends in increased patronage and better business.

*Modern Dining Room of THE NEIL HOUSE, Akron, Ohio, another "DeWitt Operated" Hotel . . . Air Conditioned—by Vilter.

● *For Efficient, Economical Air Conditioning . . . Call on Vilter.*

Superior • Since 1867
VILTER
Air Conditioning
Refrigeration . . .

The Vilter Mfg. Co.
2148 S. First Street
Milwaukee, Wisconsin

Line Co. office building, Shreveport, La.; First National Bank, Atlanta, Ga.; Federal Reserve Bank, St. Louis, Mo.; Baer-Fuller department store, St. Louis; and the Union Building, New Orleans, La., it is announced.—York Ice Machinery Corp., Roosevelt Ave., York Pa.

Wall Type Air Diffuser

No. 1789—A wall type "Anemostat" air diffuser has been developed for introducing cooled or heated air into a room. It consists of a series of diverging semi-elliptical flaring members placed one within the other, the face of the device being flush with the wall.

The standard finish of the unit is sprayed aluminum and other finishes can be furnished upon request. If it is desired that the outlet be inconspicuous, it can be purchased with the standard finish and then painted in a manner so that the outlet will merge with its surroundings.—Anemostat Corp. of America, 10 E. 39th St., New York, N. Y.

Insulating and Heat Absorbing Glass

No. 1790—The improved "Thermopane" double glass insulation was displayed at the Sixth International Heating and Ventilating Exposition in Cleveland. It comprises two panes of glass welded into one metal sealed unit, to be glazed in a single sash. For exterior use it may be had ranging from Double Strength A Quality to 1/8, 3/16 and 1/4 in. polished plate glass, with a standard 1/4 in. air space. For interior use the air space may be either 1/4 or 1/2 in. The air space between the two lights of glass is dehydrated and hermetically sealed with a metal bond at the factory to avoid dirt and condensation within it.

According to the manufacturer, this development used as window insulation provides practical insulating efficiency, reduces or eliminates room side condensation, requires no seasonal installation or removal, and has only two surfaces to be cleaned.

The same manufacturer also displayed "Aklo" heat absorbing plate glass available in standard thicknesses of 1/8 and 1/4 in. This is a plate glass of special chemical composition developed to absorb the infra-red rays from the sun. According to the manufacturer, it transmits 78 per cent of the visible light and excludes 29 per cent of the total radiant heat, as compared with 92 and 10 per cent, respectively, for ordinary window glass.—Libbey-Owens-Ford Glass Co., Box 919, Toledo, Ohio.

Heater-Ventilator Unit

No. 1791—A new "high ceiling" heater-ventilator unit, primarily suited for large area heating where machinery, cranes and other obstructions make it more convenient to mount the units overhead to blow down 15 to 50 ft was displayed at the heating and ventilating exposition in Cleveland. The unit serves the double purpose of heating in winter and ventilating or circulating air in summer; it has an adjustable damper in the top side which provides a means of changing the leaving air temperature according to the needs of the job. When opened wide with the steam shut off, it provides ventilation or circulation.

The outlet design provides four-way distribution with various velocities of discharge and the standard outlet may be adjusted or changed on the job. The heating coil is of copper fin construction. Sizes are available for 5000, 7500 and 10,000 cfm, the heating capacities being 300,000, 450,000 and 600,000 Btu per hr, respectively.—Buffalo Forge Co., 171 Mortimer St., Buffalo, N. Y.

Evaporator Pressure Regulators; Waterproof Magnetic Valves

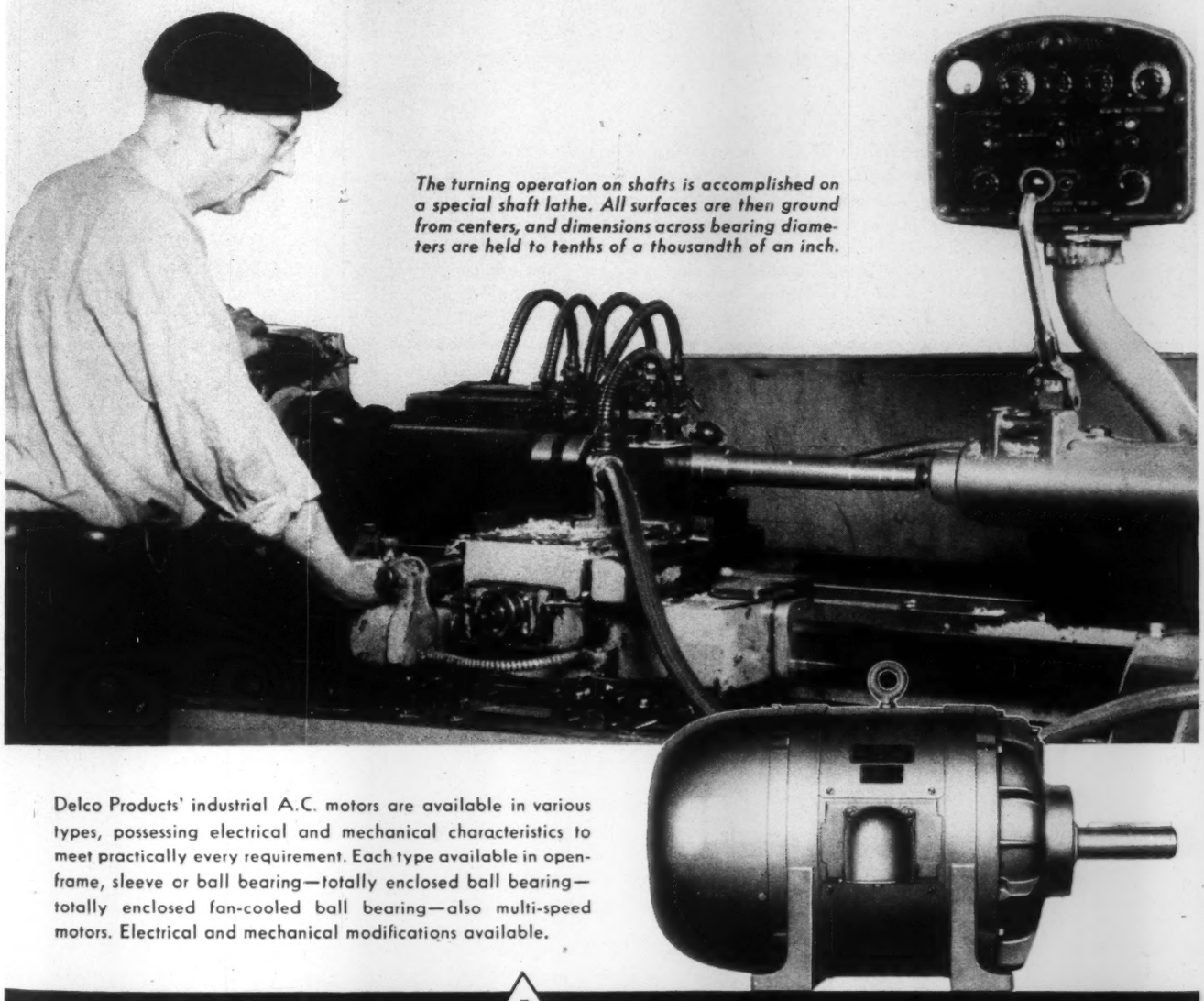
No. 1792—A new line of pilot operated evaporator pressure regulators has been announced. These regulators have been designed to maintain constant evaporator pressures regardless

Motors Designed to Do the Job

MOTORS MANUFACTURED TO DO IT WELL

What else can an industrial motor offer the user besides proper design—proper electrical and mechanical characteristics? Delco Products answers that question with its line of Delco industrial motors that give longer life—longer satisfactory performance . . . because they are manufactured to meet the most exacting General Motors standards of precision.

Many operations are accomplished on specially-built machines developed through the combined facilities of Delco Products and General Motors Research. Absolute accuracy is maintained by grouping operations, and each machined surface is registered with every other machined surface, to assure perfect alignment of the motor with the machine it is driving. Problems of mounting are further simplified by the fact that every Delco motor is dynamically balanced, to reduce vibration to a minimum. We ask you to look *within* the motor you specify—for the plus factor that you get in precision-built Delco motors.



The turning operation on shafts is accomplished on a special shaft lathe. All surfaces are then ground from centers, and dimensions across bearing diameters are held to tenths of a thousandth of an inch.

Delco Products' industrial A.C. motors are available in various types, possessing electrical and mechanical characteristics to meet practically every requirement. Each type available in open-frame, sleeve or ball bearing—totally enclosed ball bearing—totally enclosed fan-cooled ball bearing—also multi-speed motors. Electrical and mechanical modifications available.

DELCO MOTORS

DIVISION OF GENERAL MOTORS CORPORATION

DELCO PRODUCTS DAYTON, OHIO

of sudden load changes or fluctuations in suction pressure, according to the maker. They may be used on either single or multiple systems equipped with any standard liquid refrigerant control, such as float valves, thermo valves, etc.

A minimum operating differential of 2 lb between evaporator and suction pressures is sufficient for proper operation of the valve. The principle of removable valve cages has been employed in the design of the regulators. The entire valve seat and poppet assembly, i.e., the valve cage, is replaceable without removing the valve proper from the line.

The same manufacturer has also added to its line of waterproof magnetic valves the "Type S2," which is a packless, pilot operated piston type stop valve for use with "Freon," methyl chloride, brine, water, gas, air and steam. The waterproof coil and leads are designed to eliminate failure due to condensation of moisture when the valve is installed in low temperature or high humidity rooms.—Alco Valve Co., 2628 Big Bend Blvd., St. Louis, Mo.

Air Eliminator

No. 1793*—A new automatic valve, or air eliminator, may be used for removal of air from steam heated retorts, paper machines, creosoting cylinders, and "on and off" steam mains. It consists of a conventional type of diaphragm valve to which has been added a special dome shaped inlet chamber. Housed in the dome of the inlet chamber is a bellows type thermal expansion unit which raises and lowers a needle valve that seats against an orifice mounted in the top of the dome. This valve acts as a pilot valve to admit pressure to the top of the diaphragm. A pipe above the diaphragm is fitted with a small relief vent which dissipates the pressure above the diaphragm when the pilot valve is closed.

Operation is automatic and is based on the fact that air in a steam system has the effect of lowering the steam pressure and hence its temperature. The sensitivity of the air eliminator can be adjusted by means of a needle valve regulating nut. By this adjustment, it will open just a fraction of a degree below steam temperature or several degrees below, as may seem desirable for the particular service.

The unit is self-compensating for changes in steam pressure, according to the manufacturer. The unit measures 12½ in. in height, weighs 30 lb, and has valve bodies made from bronze.—Armstrong Machine Wks., 874 Maple St., Three Rivers, Mich.

V-Type Refrigeration Compressor Units

No. 1794—New refrigeration units designed primarily for air conditioning applications and utilizing a new eight cylinder, V-type compressor for "Freon-12," range in capacity from 75 to 125 tons for single units and up to 250 tons with duplex arrangement. They are built with either V-belt or direct motor drive and can be supplied with either a horizontal shell and tube type condenser mounted on the unit or without the condenser for use with a shower condenser.—Worthington Pump & Machinery Corp., Carbondale Div., Harrison, N. J.

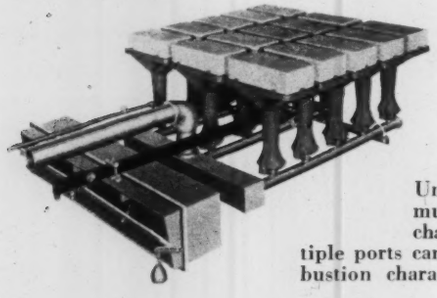
Electric Air Heaters

No. 1795.—New wall and floor mounted air heaters recently announced are particularly desirable for use in crane cabs, truck houses, valve houses, locomotive cabs, scale rooms and garages, the maker states. They are designed for use with either a three heat hand switch or a thermostat.

The wall mounted heaters range from 25⅛ to 31¾ in. in length and are from 10⅜ to 16⅞ in. high. They have a rating of 1000 to 4500 watts and operate on 115, 230 or 440 volts. The floor mounted heaters have approximately the same heating surface in inches and the same wattage rating as the wall units, but are made to operate only at 115 or 230 volts.—Westinghouse Electric & Mfg. Co., Mansfield, Ohio.

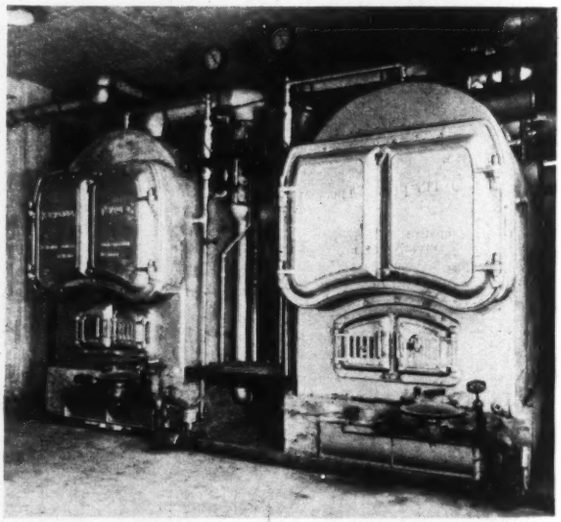
Entire School System Converted with WEBSTER BURNERS

The Series F600 WECO-N. G. E. gas burners are available for ratings from 50,000 to 10,000,000 B.t.u. output. Easily installed in any metal firebox or sectional boiler. This burner operates on straight natural gas or mixed gas to 800 B.t.u.



WECO—N.G.E. Series F600

Features — Higher capacities at low pressures; Unique baffles reduce maximum flame length; Interchangeable grills with multiple ports can be varied to suit the combustion characteristics of various gases.



The heating plant shown here is one of five in an entire school system all of which were converted to gas fuel with Series F600 WECO-N. G. E. Burners.

Write for descriptive bulletin.

THE WEBSTER ENGINEERING CO.

TULSA —DIVISION OF— **OKLAHOMA**

Surface Combustion Corp. . . . . Toledo, Ohio

SAFETY WITH *Seamless*

TEN YEARS AGO a remarkable development made its appearance in the flexible metal hose and tubing world. It was seamless metal tubing with all the flexibility of a garden hose.

Since then American Seamless Flexible Metal Tubing has found its way into thousands upon thousands of different applications, each making valuable use of one or more of its unique properties... such as:

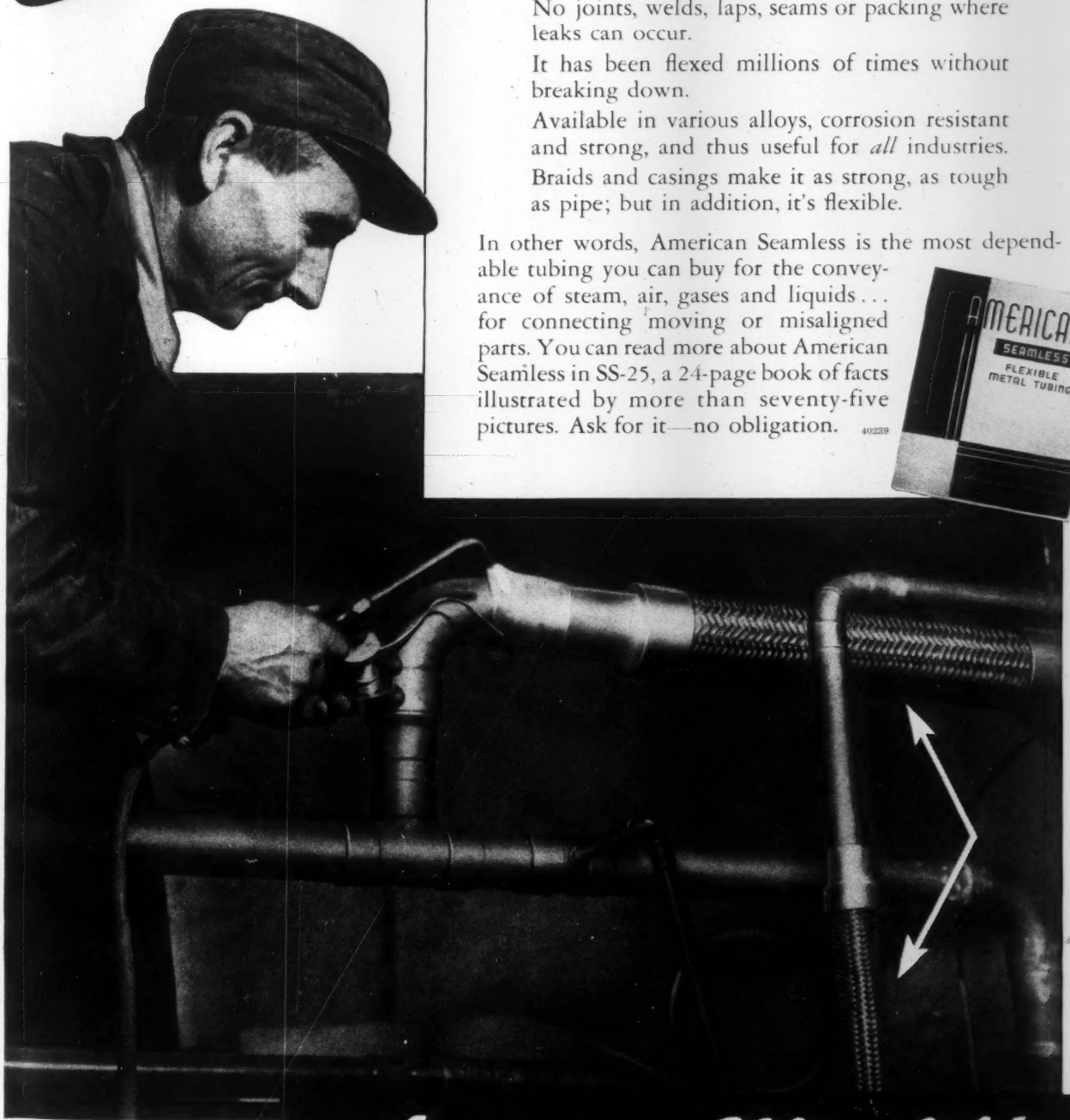
No joints, welds, laps, seams or packing where leaks can occur.

It has been flexed millions of times without breaking down.

Available in various alloys, corrosion resistant and strong, and thus useful for *all* industries.

Braids and casings make it as strong, as tough as pipe; but in addition, it's flexible.

In other words, American Seamless is the most dependable tubing you can buy for the conveyance of steam, air, gases and liquids... for connecting moving or misaligned parts. You can read more about American Seamless in SS-25, a 24-page book of facts illustrated by more than seventy-five pictures. Ask for it—no obligation. 40239



In air-conditioning systems, American Seamless eliminates vibration from compressor. See page 12 in Bulletin SS-25 for complete information.

American Metal Hose

ANACONDA
from mine to consumer

The American Metal Hose Branch of The American Brass Company
General Offices: Waterbury, Conn. *Subsidiary of Anaconda Copper Mining Company*
In Canada: Anaconda American Brass Ltd., New Toronto, Ont.

New Jet Pump

No. 1796—To bridge the gap between shallow well and deep well pumps, a jet pump unit has been developed which utilizes the ejector principle, its maker states.

This unit operates by diverting water under pressure down a drive pipe to a jet manifold. The heart of the jet manifold is a jet made of "Tenite," a plastic. The water emerges from the nozzle at a high velocity and passes upward through a venturi tube, creating a partial vacuum at this point. The vacuum in turn draws water from the well through the venturi tube creating pressure sufficient to force the water up the suction pipe to a point where the centrifugal impeller can pick it up for delivery to the pressure tank.—Crane Co., 836 S. Michigan Ave., Chicago, Ill.

Time-Temperature Controller: Two Stage Thermostats

No. 1797—The new "Chronotherm" automatic time-temperature controller is designed to operate any of the maker's Series 10 primary control and combines automatic time control with the features of the heat accelerated Series 10 thermostat. It has a self-starting telechron electric clock to lower the temperature automatically at any desired time at night and restore the normal temperature at any time during the day.

The standard operating range is 56 to 84 F, with an adjustable differential. The case is small in size for an instrument of this type (height 3⅝ in., width 5¼ in., depth 2½ in.) and is finished in rich silver bronze. It is expected that production of this instrument will equal the demand by March 15, the maker says.

The same manufacturer has also announced a new two stage thermostat for low voltage a-c application only (1 amp, 25 volts) which is designed to control the "high-low" or "high-low-off" flame sequence provided by some types of gas and oil burners. Two circuits, closing in sequence on temperature fall, are provided—one controls the high, the other the low fire valve—and both are actuated by the same bimetal element and magnet. This design prevents the device from getting out of step and assures proper operating sequence, the maker states. Standard operating range is 54 to 86 F or its Centigrade equivalent; however, other ranges are available.—Minneapolis Honeywell Regulator Co., 2701 Fourth Ave., S., Minneapolis, Minn.

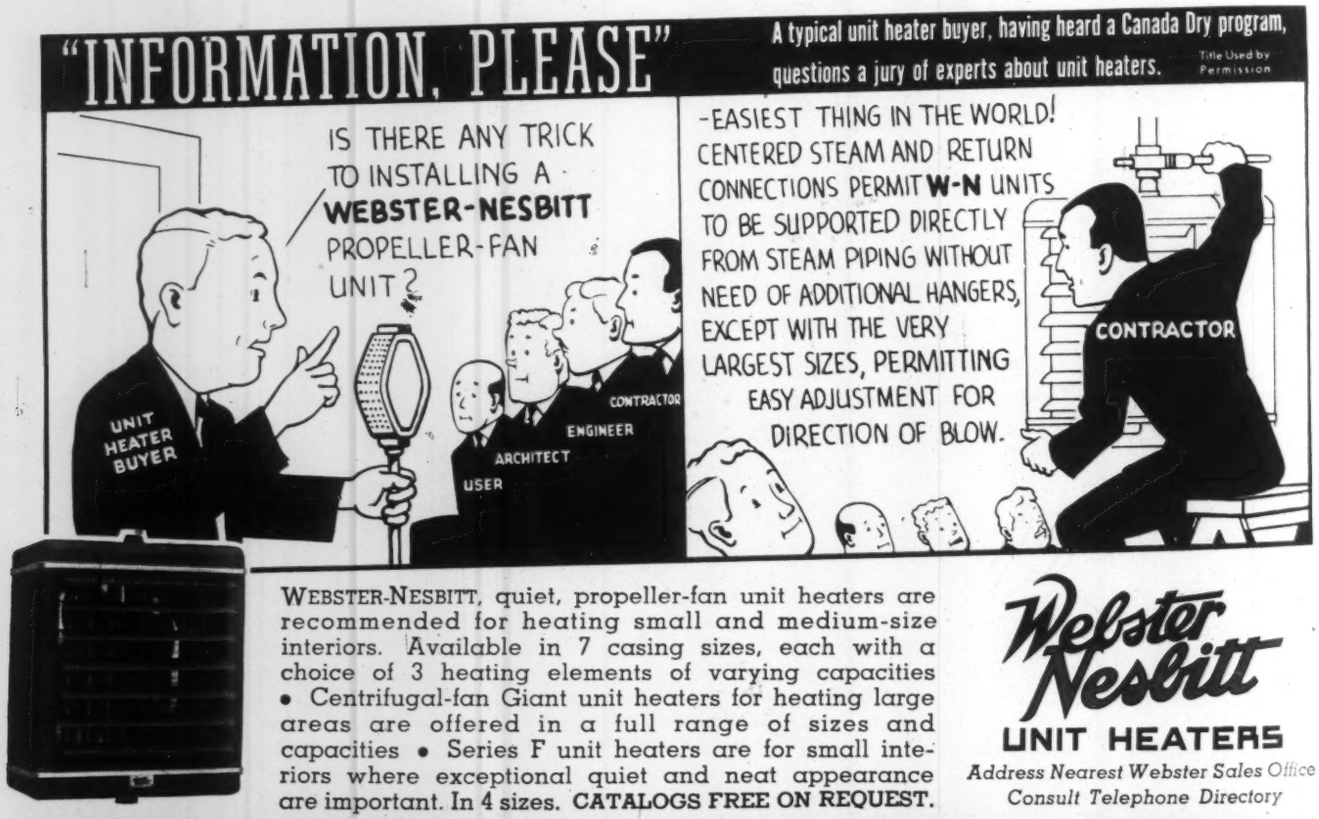
Unique Air Conditioning Patent

No. 1798—An unique method of summer air conditioning was recently patented. The basic invention is described in U. S. Reissue Patent No. 21,203 and other pending patents describe improvements.

As set forth by the patentee, the object of the system is to make use of steam or hot water heating plants to provide cooling and complete air conditioning in hot weather. A compressor in summer delivers compressed, filtered outside air at atmospheric temperatures into the piping which is used for steam or hot water heating in winter. On reaching the radiators the compressed air is cooled below zero by expanding it, and this cold air stream is mixed with another more copious stream of filtered outside air drawn in from outdoors through a window vent.

The warm outside air stream is thus cooled below the dew point, causing condensation of its moisture which is deposited within the radiator and drained through the return pipes. The cooled, dried air, after mixture with a third stream of recirculated room air, then is introduced into the room.

The expanding compressed air energizes a motor which drives the two blowers used to produce the outside and recirculated air streams; the air motor also drives a secondary air cooled compressor which acts as a booster for the main compressed air


"INFORMATION, PLEASE"
A typical unit heater buyer, having heard a Canada Dry program, questions a jury of experts about unit heaters.
Title Used by Permission
IS THERE ANY TRICK TO INSTALLING A WEBSTER-NESBITT PROPELLER-FAN UNIT?
UNIT HEATER BUYER
CONTRACTOR
ENGINEER
ARCHITECT
USER
-EASIEST THING IN THE WORLD! CENTERED STEAM AND RETURN CONNECTIONS PERMIT W-N UNITS TO BE SUPPORTED DIRECTLY FROM STEAM PIPING WITHOUT NEED OF ADDITIONAL HANGERS, EXCEPT WITH THE VERY LARGEST SIZES, PERMITTING EASY ADJUSTMENT FOR DIRECTION OF BLOW.
CONTRACTOR

WEBSTER-NESBITT, quiet, propeller-fan unit heaters are recommended for heating small and medium-size interiors. Available in 7 casing sizes, each with a choice of 3 heating elements of varying capacities • Centrifugal-fan Giant unit heaters for heating large areas are offered in a full range of sizes and capacities • Series F unit heaters are for small interiors where exceptional quiet and neat appearance are important. In 4 sizes. **CATALOGS FREE ON REQUEST.**

Webster Nesbitt
UNIT HEATERS
Address Nearest Webster Sales Office
Consult Telephone Directory

Manufactured by JOHN J. NESBITT, INC., Holmesburg, Philadelphia, Pa.
Distributed Exclusively in the U. S. A. by WARREN WEBSTER & COMPANY, Camden, N. J.

stream. These four units are mounted within the radiator enclosure on a common, vertical shaft. Temperature and humidity of the air in each room may be independently controlled, either automatically or by manual setting of two valves.—C. J. Thatcher, Consulting Engineer, 891 Park Ave., New York, N. Y.

Germ Killing Lamp

No. 1799—Investigations of a new germ killing ultra-violet lamp conducted by Dr. L. R. Koller have shown that one of the 15 watt lamps in a duct of non-reflecting material can sterilize 200 cfm of air, the maker states. The lamp is a tube of special glass filled with low pressure mercury vapor. It resembles regular fluorescent lamps except that there is no fluorescent coating inside the tube. It is available in three sizes, largest of which is the 15 watt lamp, 18 in. long and 1 in. in diameter. The lamps can be operated off the usual commercial 110 volt circuits, but require the same auxiliary equipment as fluorescent lamps.—General Electric Co., Lamp Dept., Nela Park, Cleveland, Ohio.

Liquid Level Regulation

No. 1800—Among the developments introduced at the heating and ventilating exposition in Cleveland was the new "Magnetrol" method of liquid level regulation for pump controls, low and high water cut-offs, and low and high water alarms.

Operation of the device depends upon a piston attached to the float, the piston being free to rise or fall inside a tube. Outside the tube is an "Alnico" magnet. When the piston inside the tube comes within the magnetic field of the magnet, the latter snaps forward. The action makes or breaks the circuit, depending upon the function, by changing the angle of tilt of a mercury tube switch attached to the magnet. When the float lowers (or if it rises still higher) the piston drops out of the magnetic field and the action is reversed.

With this method, the maker states, the only problem is to provide a float chamber, enclosing tube, and float ball of suitable thickness for the working pressure or temperature to be encountered.—Fred H. Schaub Engrg. Co., 325 W. Huron St., Chicago, Ill.

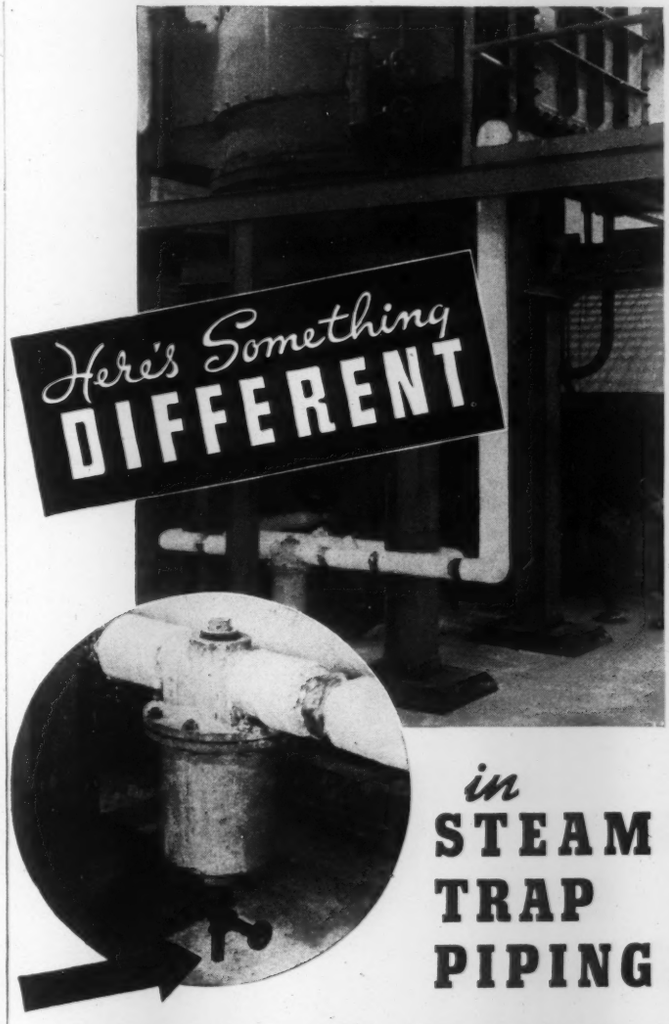
Axial Flow Fans

No. 1801—The new "Buf-flow Limit-Load" axial flow fans were on display at the recent heating and ventilating exposition in Cleveland. According to the manufacturer, the announcement completes a three year test and research period undertaken to improve (1) efficiency by correct proportioning of blades, hub, and inlet and outlet; (2) pressure characteristics by producing a non-overloading power characteristic; and (3) noise characteristics by lower speeds and directional vanes to reduce turbulence. A feature of the design is a set of stationary blades, or directional guide vanes, in the fan inlet to spiral the air in a direction opposite to wheel rotation in order that the air may be directed to leave the wheel in an axial direction.

According to the manufacturer, the power curve for the fan is quite flat and therefore motor sizes can be selected close to the actual horsepower without danger of overloading. The fan is available in an arrangement for v-belt drive, for direct drive, and for either vertical up, vertical down, or horizontal discharge. For a blowing fan a bell inlet is furnished and the exhausting unit is complete with an inlet cone flanged for connection to existing ductwork.—Buffalo Forge Co., 171 Mortimer St., Buffalo, N. Y.

Solar Heat Screens

No. 1802—Among the developments which attracted wide interest at the heating and ventilating exposition in Cleveland was the new "Kool Shade" screen, which has been designed to keep



in
STEAM TRAP PIPING

A sediment pocket and blow-down at the bottom of a steam trap *is* different. And that is a new twist that engineers at Industrial Rayon Corporation added to the advantages of simplified piping with Anderson Super-Silvertop steam traps.

Since the vertical inlet tube reaches almost to the bottom inside this steam trap, the drain plug opening just below the tube is the logical place to attach a sediment pocket and blow-down valve. Have you ever seen a more clean-cut, practical piping layout? (Trap is a No. 25 Super-Silvertop installed at the Industrial Rayon Corporation's Painesville plant.)

Another feature of the Anderson Super-Silvertop trap is guided bucket—a bucket-on-a-track confined to straight line motion—a bucket controlled in true engineering fashion to open the valve wide and close it tight. This design also permits a larger bucket in the same size case and a longer lever arm. These together open a larger valve and open it wide to give greater trap capacity.

Find out more about Super-Silvertop. Join the thousands of satisfied users who no longer put up with complicated puzzle piping and swinging bucket – write today for complete details and prices.

THE V. D. ANDERSON COMPANY
1949 WEST 96th STREET • CLEVELAND, OHIO

SUPER-SILVERTOP STEAM TRAPS

sun heat out of a building. It is made of horizontal flat bronze strips, woven together every half-inch vertically with bronze wire, these flat strips functioning like a partly open venetian blind. These strips are no further apart than the wires of ordinary insect screening, and the screen keeps out insects as well as reducing solar heat gain.

According to the manufacturer, tests have indicated that this screen transmits only 19 per cent of the total solar heat reaching it. Thus it is of advantage in enabling reduction in capacity of the air conditioning equipment for a building, in reducing operating costs or enabling the conditioning of a larger space without increasing the capacity of the plant.—Ingersoll Steel & Disc Co., 310 S. Michigan Ave., Chicago, Ill.

Stack Thermometer

No. 1803—With a new all-metal stack thermometer introduced recently, flue gas temperatures can be measured by fitting it into a sleeve holder furnished with the unit. The sleeve holds the thermometer stem in a horizontal position in the center of the flue gas stream without danger of dropping it into the flue.

Completely enclosing the temperature sensitive element, the stem is 9 in. in length. A shatterproof scale glass is employed. The temperature range is from 200 to 1000 F, and accuracy is guaranteed within 1 per cent by the maker.—Weston Electrical Instrument Co., 619 Frelinghuysen Ave., Newark, N. J.

Bronze Valves

No. 1804—A new line of bronze globe and angle valves with plug type discs and renewable seat rings is announced for close control in throttling service and for general heavy duty. These valves are made in sizes from ¼ to 2 in. for 200 lb steam at 550 F and 400 lb cold water, oil or gas, non-shock; and in sizes from ¼ to 3 in. for 300 lb steam at 550 F and 600 lb cold water, oil or gas, non-shock.

The stems have 60,000 lb per sq in. tensile strength with [illegible] standard threads, and have rounded stem head to permit self-centering of the disc. The bonnets are provided with heavy union bonnet rings, and have a machined seat on the under surface to permit repacking under full line pressure when the valve is wide open.—Kennedy Valve Mfg. Co., Elmira, N. Y.

Pressure and Low Water Control

No. 1805*—A new combination pressure and low water control is offered for the protection of low pressure, automatically fired steam boilers against the hazard of firing into a dry boiler or building up excess pressure. The control is furnished standard with quick hook-up attachments for simplified installation complying with the ASME code, according to the maker. It is of packless construction, the seal being a metal bellows.

Double outside adjustments are provided for setting the pressure range, which is adjustable from 0 to 10 lb. Electrical capacity is 10 amp 115 volts; 5 amp 230 volts. (Motor rating 1 hp repulsion-induction, 1 hp split phase or d-c.)—Mercoid Corp., 4201 Belmont Ave., Chicago, Ill.

Centrifugal Pump Unit

No. 1806—A new balanced "Monobloc" centrifugal pump wherein (according to the maker) the motor and pump have been engineered as an integral unit of balanced functional design has been announced. It is compact, requiring minimum floor space, but is provided with ample room for repacking the stuffing box. A shaft of large diameter, on rigid bearing mountings, maintains concentricity of rotating parts to increase packing life. The new drip-proof motor features a directed flow of ventilating air to prevent drawing of moisture into the motor.

This unit is offered in sizes from 1 to 4 in. with capacities to

Why Continue To Suffer From Summer's Heat?

With Young evaporators and cooling coils you have available all the essential cooling surfaces to do any air conditioning or cooling job, using Freon, methyl chloride, water or brine as the coolant.

From the Young catalog you can select a coil of the proper dimension and capacity to meet your specifications.

The Young representative in your vicinity is a competent engineer who will gladly work out your cooling problems without cost or obligation. Now is the time to plan to keep cool this summer. Write for literature on cooling.

YOUNG RADIATOR COMPANY
RACINE WISCONSIN

Young Quality

1000 gpm against heads up to 280 ft. It is available in bronze fitted, all-iron and all-bronze construction, horizontal or vertical mountings, with motor of any electrical characteristics.—Worthington Pump & Machinery Corp., Harrison, N. J.

Refrigeration Compressor Control

No. 1807—The new light duty "Polartrons" may be used to control refrigeration compressor motors under 1 hp, or for larger sizes if used to pilot a magnetic starter. They embody the features of the present heavy duty type, including "frost free constant cold control" of refrigeration coils and independent adjustment of on and off suction pressure, the maker points out.

Models are available for low side or both low and high side pressure control. A small pilot relay inserted in the case may be attached in the field, if desired, to permit room thermostat control of refrigeration.

The switching mechanism is a newly designed, snap action, non-magnetic switch. Separate adjusting dials (calibrated if desired) are supplied for on and off suction pressures. A special auxiliary "cooling control lever" is furnished which, when attached, will permit limited regulation of the temperature by the operator.—Minneapolis-Honeywell Regulator Co., 2701 Fourth Ave., South, Minneapolis, Minn.

Protects Refrigeration Valves

No. 1808—A development for increasing operating efficiency of expansion or solenoid valves in refrigeration systems is the new "Trap-it" strainer or filter unit recently announced. It is designed to trap impurities such as water, scale, gummy deposits and particles of solder. It contains nothing that can dissolve or be injurious to the refrigeration system, the manufacturer reports, and it prevents many of the causes for sticking or slow acting valves which can often be traced to sources other than the valve itself.

At the present time the unit is available in three sizes. The small size is furnished with a ¼ or ⅜ in. SAE male flare inlet and outlet connection and is suitable for installations of 1 ton. The medium size, also available with the ¼ or ⅜ in. inlet and outlet connections, will handle 2 tons of refrigeration, while the large size unit is adequate for systems up to 4 tons, and is available with ⅜ or ½ in. inlet and outlet connections. These recommendations for size are based on the use of "Freon." The unit will handle twice the amount of refrigeration where methyl chloride or sulphur dioxide is used.

The filtering element is a specially wound bobbin of coarse processed cotton yarn of soft texture.—Automatic Products Co., 2460 N. 32nd St., Milwaukee, Wis.

Self-Contained Room Coolers

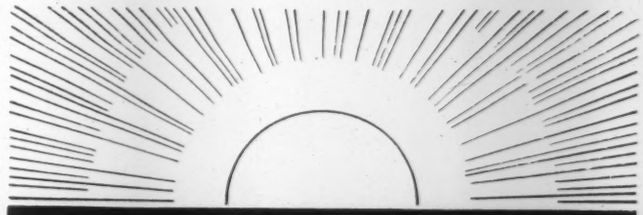
No. 1809—Three sizes of "Mobilaire" packaged room coolers requiring no plumbing connections, are available with capacities of 4000, 6000, and 8500 Btu per hr. The new line includes two models of the type that are installed directly into the window frame and project out a few inches into the room.

The third unit is a floor model and has a rated delivery of 300 cfm conditioned air.—Westinghouse Electric & Mfg. Co., East Springfield, Mass.

Improved Line of Control Valves

No. 1810—An improved line of "Stabilflo" controlled valves designed for accurate response to the demands of a control instrument has been announced.

As equipped with a wide range V-port plunger, the valve has a rangeability of 50 to 1, the maker states. When this valve is lifted off the seats, the initial clearance flow is 2 per cent. From this 2 per cent flow to 100 per cent flow, the valve gives equal



THERE IS SOMETHING NEW UNDER THE SUN!

Make this test: Try to jab a pin or piece of wire *straight* through the AIR-MAZE air filter panel. You'll hit a baffle as soon as you get through the first screen. Yet air goes through freely with practically no restriction.

Type A AIR-MAZE Air Filter Panel.

The reason is that AIR-MAZE construction embodies a unique and scientific principle which is *radically distinctive* in its application to air filter design. AIR-MAZE filter panels consist of a maze of alternately placed and exactly spaced crimped galvanized wire screens. These screens are not only graduated in size from large crimps to small but are so alternately placed as to form positive baffles which intercept the passage of even the finest dust, pollen and lint.

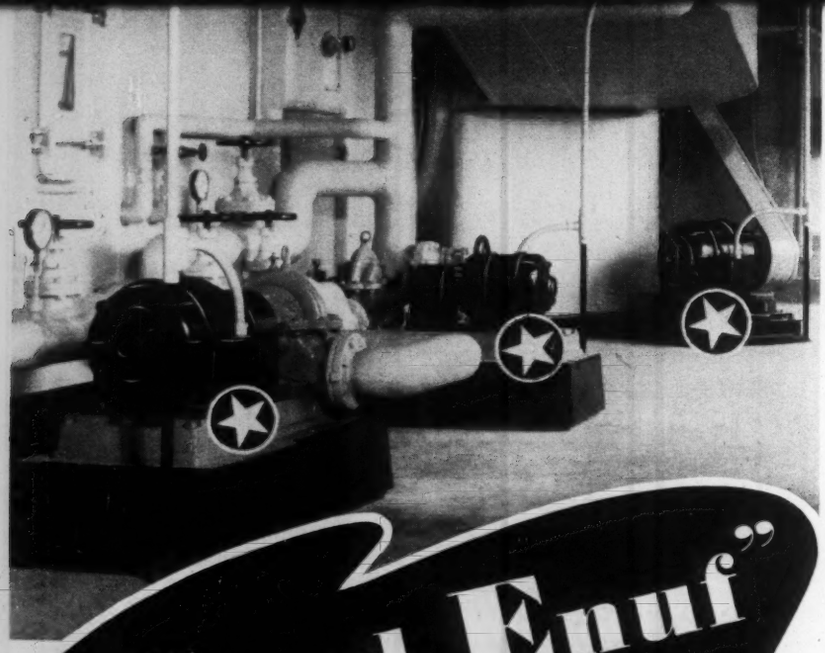
This novel baffle principle causes the flow of air to change direction and be divided again and again during its passage through the filter panel. In this manner entrained particles in the air cannot escape impingement on the evenly spaced viscous coated baffles.

AIR-MAZE filters are economical because they need no replacement and cost only *a few cents* to clean them for renewed, efficient, trouble-free performance. Of all metal construction they are *approved by the Underwriters Laboratories as fire retardant.* They are odorless, sturdy and easy to handle as they slip readily into existing panel spaces. Write for detailed information.

APPROVED BY THE UNDERWRITERS LABORATORIES (UNDERWRITERS' APPROVED SERVICING METHOD NECESSARY)

AIR-MAZE CORPORATION
5200 Harvard Avenue Cleveland, Ohio

AIR-MAZE CLEANABLE AIR FILTER PANELS



"Good Enuf" is NOT the Way HOWELL Operates

LONG on Performance-SHORT on Decibels!

Type SC Air Conditioning Motors—sizes, ½ to 150 H.P.

For many years we have specialized in the air conditioning field, handling a large volume of business annually.

Manufacturers, engineers, contractors *and users* learned long ago the VALUE of using HOWELL Air Conditioning Motors.

They found that these motors measure up to any requirement, no matter how exacting . . . run EXTREMELY QUIET . . . that ratings are liberal . . . service prompt and reliable.

"Good enuf" has never been a policy at HOWELL. Since the first HOWELL Motor was wound in 1915, we have constantly sought — *and attained* — product improvement.

In short, when you specify and use HOWELL Air Conditioning Motors, you are protecting your installation in the best possible manner against *any motor trouble whatsoever.*

We suggest that our Bulletin SC will interest you. Write for it!

1915 · 1940 HOWELL MOTORS 25th Anniversary Year

★*Illustration at top shows HOWELL Motors driving air washer pumps and main supply fan in office building air conditioning system of large pharmaceutical manufacturer. This company INSISTS on HOWELL, as a result of over 20 years' experience with hundreds of HOWELL Motors in its various plants.*

HOWELL ELECTRIC MOTORS COMPANY
HOWELL, MICHIGAN
Representatives In All Principal Cities

percentage increases in flow for equal amounts of valve l[illegible] according to the announcement. The V-port plunger is s[illegible]wed and pinned to the valve stem and designed for easy field re[illegible]sal. In addition to this V-port valve plunger, two other typ[illegible] are available for exceptional control applications.

Operation of the valves is provided by a floating power [illegible]tor which incorporates suspended construction to eliminate nee[illegible] for guides and anti-friction bearings. The motor may be re[illegible]oved without disturbing the motor adjustments or the relation be[illegible]ween valve and motor.—Foxboro Co., 106 Neponset Ave., Fo[illegible]oro, Mass.

Plug Cock for Gage Lines

No. 1811—A lubricated gage cock which can be completely opened or closed with a one-fourth turn has been developed for test pressures up to 4000 lb and with a rated working pressure for water, oil and gas of 2000 lb.

Its body is forged steel and it has a stainless steel plug. Stick lubricant is inserted under a lubricant screw which can be turned down. Turning of the screw transmits hydraulic pressure to the seat, and in the event the cock becomes hard to turn, the pressure exerted will loosen it. Special lubricants are available for steam.

This gage cock is made in ¼ and ⅜ in. sizes and is wrench operated.—Merco Nordstrom Valve Co., 400 N. Lexington Ave., Pittsburgh, Pa.

Self-Contained Air Conditioners

No. 1812—The "Yorkaire" line of self-contained air conditioners for commercial establishments has been expanded and improved to include units from 2 to 15 hp, it was announced last month.

The new model No. 225 (standing slightly over 5 ft high), and Nos. 275, 350, 475, and 550, are used in small to medium sized business places, where the conditioner is within the sales area, and are housed in upright cabinets requiring little floor space. They can be used with or without ducts for air distribution. The models numbered 700, 1000 and 1500 are used in medium to large sized establishments where the conditioner is concealed in the basement or anteroom and air is distributed by means of duct work. These larger conditioners are sectional in design and can be moved in and assembled to form a single complete unit.—York Ice Machinery Corp., Roosevelt Ave., York, Pa.

New Flaring Tool

No. 1813—An improved tool for flaring copper, brass and aluminum tubing is designed for use in making up standard SAE flare connections. A feature of the tool is its yoke, which is made so that it can be slipped directly over the bar without twisting or turning, the maker states. This has been accomplished by slotting the inside edge of the yoke so that, once in position, a slight turn holds it in place.

The construction of the bottom of the yoke makes it possible to make flares even when there is little space between flare nut and end of tubing. This is said to be particularly advantageous with fin tubing. The yoke is of forged steel construction.

Two sizes of the tool are offered, one for handling ⅜, ¼, ⅝, ⅜, ⅞ and ½ in. OD tubing and the other for handling ¼, ⅜, ⅝, ½ and ⅝ in. tubing.—Imperial Brass Mfg. Co., 1200 W. Harrison St., Chicago, Ill.

High Speed Synchronous Motor

No. 1814—A new high speed synchronous motor of "packaged" construction has been developed with the aim of simplifying installation and wiring and reducing space requirements. Wiring on the job for the motor is the same as for a squirrel cage in-

duction motor, need for a separate, panel type control having been eliminated, the maker states.

The motor consists of a fabricated, all steel motor of compact design; an exciter, mounted on top of and V-belt driven by the motor, or direct connected to the motor; and a field control unit mounted on the motor, for automatically applying and removing excitation for the field winding of the synchronous motor. The exciter and field control unit are wired to the motor at the factory. Thus, the only wiring required on the job is between the synchronous motor and the line control unit which can be an ordinary, wall mounted, across the line or compensator type induction motor starter.

These motors are available up to 350 hp at 1800 rpm.—Electric Machinery Mfg. Co., 1331 Tyler St., N. E., Minneapolis, Minn.

Gage for High Pressure Oxygen

No. 1815—A new special oxygen "Duragauge" for high pressure oxygen service has been announced. The case is a heavy brass casting with a "safety solid front" and a full open back. The back of the gage is sealed with moistureproof paper protected by a thin brass disc. The mounting lugs on the case hold it away from the wall.

The gage cover is of transparent plastic to open up the dial for easier reading.

Accuracy is guaranteed to within ½ of 1 per cent, the maker states.—Manning, Maxwell & Moore, Inc., Ashcroft American Gauge Div., Bridgeport, Conn.

Improved Underfeed Stokers

No. 1816—Improved models of the "Vulcan" and "Lo-Set" ram underfeed stokers were announced and displayed at the recent International Heating and Ventilating Exposition in Cleveland. These stokers are for heating and power plants and now offer —states the manufacturer— improved retort design, a new method of combustion control, and a mechanical drive guaranteed for five years.

Combustion control regulates the coal feed, air supply and overfire draft in ratio with the steam demand by means of timing devices; no variable speed or double shaft motors are required, according to the manufacturer.—Canton Stoker Corp., 407 Andrew Pl., S.W., Canton, Ohio.

Flexible Bronze Tubing

No. 1817—A new 4 in. ID flexible seamless bronze tubing has been developed. For frequent flexing a minimum bending radius of 24 in. is recommended by its maker, although a length of the tubing of about 5 ft can be bent by hand to a radius of but 6 in.

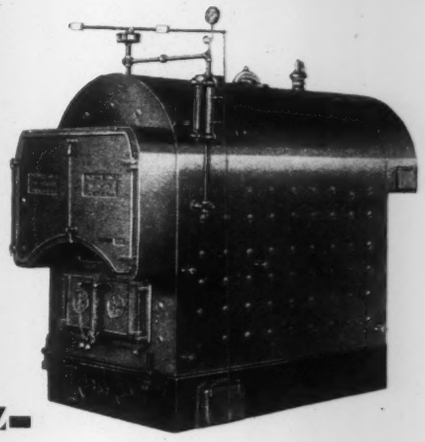
According to the announcement, pronounced S-shaped corrugations ¾ in. deep and relatively close pitch are responsible for the flexibility. The convolutions form a lefthand helical thread to assure uniform distribution of stress and long life of the hose.

Covered with bronze braid jacket, this tubing will withstand a safe internal working pressure of 150 lb per sq in. and is suitable for conveying steam, oil, water, gas, air, etc., states the maker. The principal applications include vibration elimination, diesel exhaust tubing, etc. Weight per foot is approximately 6.5 lb. Suitable pipe thread couplings or flanges are available in standard sizes.—Seamlex Co., Inc., 5-19 48th Ave., Long Island City, N. Y.

Automatic Reset Timers

No. 1818—A new series of automatic reset timers and time delay relays are especially adapted to provide a variable time delay between the operations of commercial or industrial stokers

Burnham Welded Steel is made in 19 sizes ranging in capacities from 1,800 to 42,500 sq. ft.



Hank SEZ—

What's In Front of a Boiler Is What's In Back of It

HERE lately I went inter a drug store with a friend what needed a tube of some shaving suds.

There wuz a smart squirt behind the counter what tried to sell him one of them wild cat brands costing a lot less.

But my friend insisted on Colgate's.

When we got outside I said, "What the heck, Bill, why not save the money."

Bill replied, "I don't know no one in Colgate's personally, or what's in their shaving suds, but I know the reputation Colgate has, and that's enuff fur me to go by."

"Same with steel boilers," sez he.

"They're all built to a specification. But in addition, Burnham's is backed by a reputation.

"One built up in 73 years of making whatever they made, jist the best they knowed how. Which is good enuff fur most folks."

Hank Hindle

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Today, the Dependability, Accuracy, and Supersensitivity of A-P Valves are so widely known and consistently PROVEN that the thought of a substitute is impossible!
In stock at Progressive Jobbers everywhere . . . . Recommended by Progressive Service Men . . . A-P manufactures a complete line of Expansion Valves, Water Valves, Solenoids, Oil Valves, Temperature Control Sets, and "Trap-It," a new type of filter for all Refrigeration Systems.
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or oil burners directly controlled by pressure, the manufacturer states. The following uses suggest the applications of these instruments: (1) to close and then re-open a circuit; (2) to open and then re-close a circuit; (3) to make a momentary contact; (4) to repeat a pre-set schedule of momentary contacts or timed on and off operations; and (5) to reset instantaneously after a power failure.

These timers are furnished with eight terminals. A lock, magnetically operated, frees the timers of any disconnection due to vibration, the announcement states.

The same manufacturer has also recently developed a new seven day time switch to permit any variable arrangement of on and off operations adjusted as close as three hours apart, every day for a week. Trippers are in the form of riders attached to any spot in the rim of the dial by a small thumb nut. This permits their placement so that, if desired, the operating schedule for every day in the week may be different; any day or days may be omitted, or as many as eight to 10 operations in one day can be set. Applications include stoker and oil burner control, ventilation systems, mine pumping systems, lighting schedules, etc.—Paragon Electric Co., 37 W. Van Buren St., Chicago, Ill.

New Spray Nozzles

No. 1819—A new spray nozzle gives a spray of a flat type with a slightly heavy center and can be had in a number of different spray angles. Capacities range from 1.5 to 3.5 gpm at 10 lb pressure, and 3 to 7 gpm at 40 lb pressure. Standard stock construction is brass or all iron, but other metals can be specified. The maker recommends its use for applications where spray patterns are overlapping, requiring a flat spray with a heavy center.

The same manufacturer has also announced a pneumatic atomizing nozzle available with internally mixed round or flat spray of standard brass construction, although other metals can be specified. It is recommended for chemical processes and humidifying and industrial work where a constant proportion is to be maintained for intermittent spraying.—Spraying Systems Co., 4021 W. Lake St., Chicago, Ill.

Pocket CO_2 Indicator

No. 1820—An addition to the maker's line of combustion testing instruments is a new pocket CO_2 indicator. Its features, according to the announcement, include unbreakable transparent plastic construction, a carrying case with built-in compartments for draft gage and thermometer, and self-closing valves to eliminate possibility of losing the absorbent solution.

Making the rubber tubing connection is the only set-up required and analyses are run rapidly by holding the inlet plunger down while pumping the sample in and raising and lowering the absorption unit, the maker says.—F. W. Dwyer Mfg. Co., 565 W. Washington Blvd., Chicago, Ill.

V-Belt Drive Blowers

No. 1821—A new v-belt drive unit blower has been developed to meet the need for a combined unit of this type. It is especially adapted—says the maker—for installations where on account of local conditions, or due to the preference of the architect, owner or contractor, a v-belt drive is considered more desirable than a direct connected blower.

The arrangement comprises the blower, which is driven by a standard frame motor through a drive, the motor being mounted on a steel pedestal base built integrally with the blower housing. Flexibility of operation is a feature, as is its universal discharge which permits changing the nozzle position when necessary.

Ohio Firm of Analytical and Consulting Engineers Reports on a Rhoads TANNATE Installation in an ice plant:

"When we were engaged as consulting engineers for this ice company about a year ago, we found compressors operating at 98 RPM and driven with two 75 HP Induction motors. On account of the very close shaft centers, they were using two very large, inefficient belt tighteners, resulting in a 15 to 17 per cent power transmission loss.

"Our first decision was to increase the speed of the compressors from 98 RPM to 164 RPM. This increased speed built up a load requiring 100 HP motors. Our decision to use synchronous motors is obvious.

"In our original design, we had planned to use V-belt drives. This plan was adopted on account of the very close shaft centers, however, when we learned the costs involved, to remove these very large and heavy flywheels, we were prompted to make serious investigation.

"The Rockwood Base with Tannate Flat Belt drive was brought to our attention. After a very careful study of the principles and characteristics of this type of drive, we decided to use the Rockwood Base with Tannate Flat Belt. This being our first installation with this type of power transmission, we did have some doubts as to the outcome.

"The change was made, the installation of both machines completed, both of them running through this past summer season and we feel justly proud of our engineering on this job and of the equipment used, because as a result of this change, we have increased the capacity of this plant from 60 tons per day to 80 tons per day. In addition to this, all ice produced shows a saving of 13.8 cents per ton.

"We are now convinced that at least four cents of this saving per ton is due to the perfect installation of the Rockwood Base and Tannate Flat Belt drive. We have absolutely no transmission losses whatever."

They conclude: "We are so proud of this installation that we invite anyone interested in perfect power transmission to visit this plant and witness its operation!"

That was 4 years ago—and the story is still the same!

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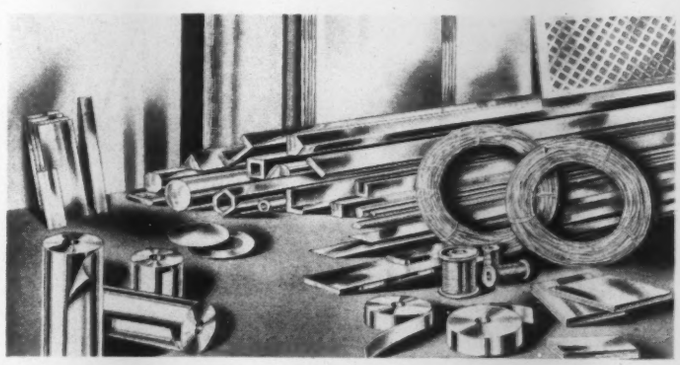
C. G. HUSSEY & CO.



Copper **HUSSEY** Brass

(Division of Copper Range Co.)
ROLLING MILLS AND GENERAL OFFICES: PITTSBURGH, PA.
Warehouses in Principal Cities

These units are available in wheel diameters ranging from 12½ to 25 in.—Autovent Fan & Blower Co., 1811-27 N. Kostner Ave., Chicago, Ill.

Portable Return Tubular Boilers

No. 1822—Announcement has recently been made of the "Power-Pak" unit portable return tubular boilers for heating and power. The entire unit—shell, tubes, furnace and refractory—is completely assembled by the manufacturer and shipped as a single package. Skids are provided so that installation work comprises sliding the boiler into position over its base and making the final connections, the maker states. Openings are fixed to accommodate the correct size of "Victor" hydraulic stokers.

These boilers are built for 100 and 125 lb working pressure and are available in six units rated at 30, 35, 39, 44, 52 and 61 hp, or 4250, 4930, 5440, 6120, 7300 and 8500-sq ft steam radiation output.

In addition to being stoker fired, these boiler units may be oil, gas or hand fired.—Orr & Sembower, Inc., Box 817, Reading, Pa.

New Steel Making Process

No. 1823—Announcement of the new "Pluramelt" steel making process has been withhheld, according to the manufacturer, until it could be said that the product was immediately available in some commercial form. In the oil refinery industry, however, pressure vessels made by this means have been in service for over two years.

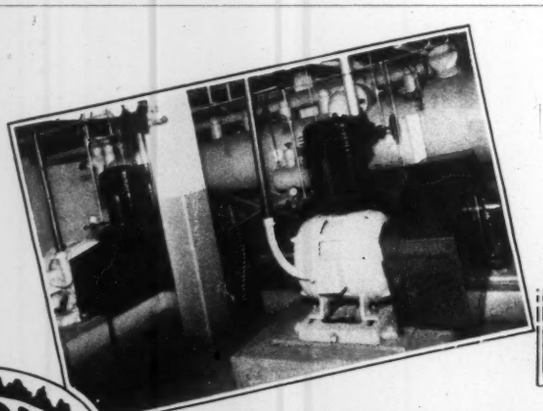
Briefly, the result of the process is to make possible more than one composition or analysis in a single final ingot. The chemical, petroleum, food and beverage, paper and other industries use vessels that require strength against the pressures carried and resistance to corrosive liquids or gases; high pressure sometimes demands wall thicknesses of 2 in. or more, yet only the inner surface must resist the corrosive contents. There has thus been a great deal of work on combining stainless steels with the cheaper carbon steels to meet such requirements.

One group of metallurgical researchers decided that the desired result could be accomplished by development of a new melting technique and after intensive investigation the result was achieved by means of a special type of electric arc melting furnace so designed that the melting operation could be carried on and the final product would be intermelted with any other metal part and allowed to solidify as an inseparable portion of the unit mass. The work culminated in the construction of a plant at Brackenridge, Pa., which is now in commercial operation.

Experience so far has been limited to "Pluramelts" of stainless alloys (high chromium-iron and chromium-nickel-iron alloys) on soft or medium steel, and tool steel types on soft steel bases, states the maker, but it seems likely that the commercial success with these materials may be duplicated with any other alloy that can be melted in an arc furnace. High carbon steel can be added to mild steel (or vice versa), the low alloy SAE steels used in any desired combination, and the high nickel alloys can be so treated. The base metal can be covered on both sides and triple or more layers made of any desired proportionate thicknesses, it is stated.

According to the manufacturer, "Pluramelt" in its numerous analyses is the product of an electric furnace melting technique, as the result of which all of the special composition materials and a small part of the low cost materials are melted and integrally joined during this operation. In this way special steels are melted and joined to low cost steels in ingot form.

It is stated that ample shop experience indicates that the product lends itself well to shop practices and can be fabricated in the same ways used for fabricating other metals.—Allegheny Ludlum Steel Corp., Pittsburgh, Pa.



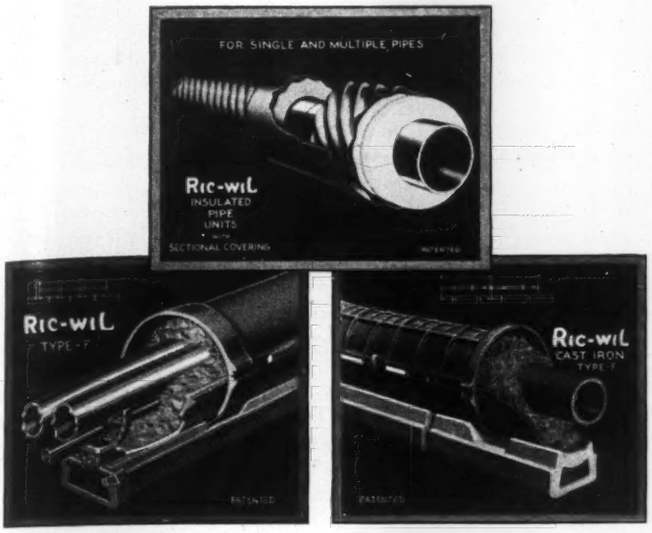
FRICK TRADE MARK **Air Conditioning**

Keeps cool and comfortable this $750,000 Supreme Court Building in Nashville, Tenn. Two 8¾ by 6 Frick Freon-12 compressors, with Frick condensers and coolers, carry the refrigerating load.

Thousands of Frick plants are giving dependable cooling service for air conditioning, ice-making and cold storage throughout the world.

You too can enjoy the benefits made available through Frick Refrigeration. Get in touch with your nearest Frick Branch or Distributor today.

Frick Co., Waynesboro, Penna.



What type of Underground Conduit Is Best for You?

In the six basic types of Ric-wiL Underground Steam Conduit you find the most complete line on the market today. These designs afford a choice of all *approved* materials—tile, heavy-duty tile, Cast Iron, and Armco Hel-Cor Ingot Iron—and also a variety of insulations including the famous Dry-paC Waterproof Asbestos, standard sectional pipe coverings, and bulk commercial insulations.

All these Ric-wiL Systems are in successful use under actual service conditions. Underground steam protection has been Ric-wiL's *exclusive business* for a third of a century. Ric-wiL's engineering technique is unchallenged.

Now—with the most complete line available *anywhere*—Ric-wiL offers a range of materials and design which effectively meets *any* requirements of cost, function, soil, surface traffic, or labor man-hours.

One end in underground steam transmission is permanent efficiency. Another is economy—first and last. Ric-wiL can help you make *both ends meet*—and will, if you say so.

Write for Catalog showing all types of Ric-wiL Conduit.

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Ric-wiL

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Booklets, Reports and Papers

Lighting and Air Conditioning

Howard M. Sharp, manager of the lighting bureau of Buffalo, Niagara & Eastern Power Corp., discussed recent developments in lighting and air conditioning at the January meeting of the Chicago section of the Illuminating Engineering Society. The following paragraphs have been taken from his talk:

Lighting and air conditioning are probably two of the most important contributions to man's continual striving for a better, more pleasing, and healthful environment. In some particulars, each moves within independent spheres, but in others there are many inter-related factors that require consideration. Ten years ago, neither of these two contributions presented any serious problems other than those encountered within the sphere of each in technical advancement and market development. Air conditioning was definitely in the luxury class, confined generally to large interiors devoted usually to entertainment. In the realm of lighting, the concern was chiefly with providing a not very adequate substitute for daylight, limited quite generally to 10 foot-candles, with resulting electrical loads of approximately two watts per square foot. The situation today is vastly different, due to the remarkable strides that have been made in each field. New knowledge of lighting has not only changed the concept of what it is and what it can do but the market has been tremendously expanded. Likewise, technical developments and the manifold commercial advantages have caused rapid strides in the promotion and acceptance of air conditioning.

The two greatest responses of an individual to an environment are through the eyes and through the skin. Within the first few weeks of the development of the human embryo, two offshoots of brain cell structure reach out from the central nerve area to the skin, and from the point of contact the marvelous structure of the eye ultimately develops. Fundamentally, then, eyes and skin start from the same family of cells. It is psychologically interesting to know that many sensations which come through the eyes set up mental reactions that tend to associate themselves with sensations coming through the skin.

This is particularly true in the realm of color. For example, strong sunshine carries with it great amounts of heat; sunshine is predominantly yellow, so that yellow and its associated hues and tints are known as warm colors. Those colors even when devoid of heat still tend to make an individual experience a warm sensation, and likewise strong lighting, even though devoid of heat, but yellow in color, may set up sensations of warmth. When we indulge in swimming, the color of water is blue or green, and it is cool to the skin. We therefore associate those colors with sensations of coolness. Likewise, the green grass is cool to the touch, due to rapid interchange of heat between the skin and the earth, and, presto, green becomes a cool color.

I have known of interiors where changes in the color of lighting from the yellows toward the blue-green have produced sensations of coolness even though the sensible heat gain in the room was not altered.

One kilowatthour of electric energy either in lighting or in appliances will produce 3413 Btu. Ordinary incandescent lighting in commercial usage is composed of approximately 11 per cent light energy, 78 per cent radiant energy, and 11 per cent in a form which is dissipated by filaments, supports, and the gas—probably being quickly translated into sensible heat. Rather careful investigations lead to the conclusion that not all of this energy becomes load on the cooling system, because some of the radiant energy passes through windows or other openings, and

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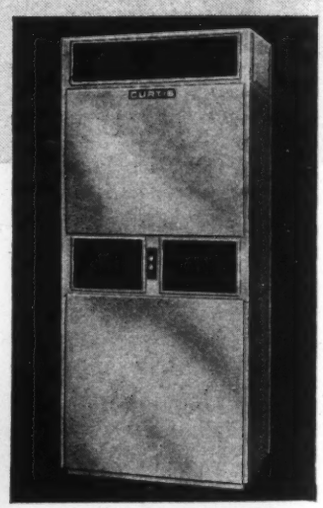

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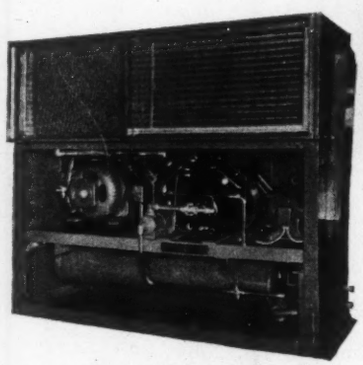

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some is absorbed by walls and ceilings, and re-radiated outside the area. Furthermore, not all of the connected lighting load is always used coincidentally with the air conditioning. However, if the designers assume that whatever connected lighting load is in use is contributing a full quantity of Btu's, it is evident that a comfortable factor of safety has been introduced.

From the standpoint of measurable sensible heat, investigation has shown that the contribution of lighting is not as great as is often suspected. Sturrock and Walker report findings as shown in Table 1.

Table 1—Temperature Rise in Degrees Fahrenheit in 17 Ft x 17 Ft Office, 11 Ft 9 In. Ceiling, over 7 Hr Period. Summer Weather, Natural Ventilation. One Door, Two Windows

Watts per Sq Ft Lighting	*Openings Closed Temp. Rise*	*Transom and Windows Open Temp. Rise*
1	1° F	¼° F
2	1.5	½
3	2.25	¾
4	3	1
5	4	1.25
6	4.3	1.5
8	6	1.75
10	7.5	2.25
12	9	2.75
16	12	3.5
20	15	4.5

From Sturrock and Walker, "The Heating Effect of Artificial Lighting," IES *Transactions*, Vol. 29, No. 1.

Some interesting exploratory investigations have been made in an attempt to determine the radiation and convection temperature contribution of lighting systems and to then translate that temperature into equivalent temperatures by means of which comfort sensations might be predicted. These studies show that for indirect lighting systems of low brightness, wattages per square foot as high as eight contribute at the most around 2 F of equivalent temperature. At the same time, reactions of occupants to these areas has shown that in many instances much lower equivalent temperature contributions are considered by occupants of the area to be causing discomfort. Further measurements have disclosed under these circumstances that the equivalent temperature of the area, either with or without lights, was greatly exceeding the average comfort zone temperature for the season of the year. In fact, a number of areas disclosed equivalent temperatures in a range where previous investigations have shown that almost 100 per cent of the occupants would experience discomfort. Under these conditions it is possible for even a modest contribution of equivalent temperature to increase discomfort and, because of the psychological tendency to transpose skin discomfort to visual sensations, the lighting systems are erroneously blamed for the discomfort. This is borne out by investigating equivalent temperatures, with and without lighting, in areas where no discomfort has been claimed by occupants and finding that the equivalent temperatures were about at the median of the comfort zones. This has been found to be true even where the equivalent temperature contribution of the lighting system was around 2 F.

Considerable work has been done by way of ventilating lighting equipment and withdrawing thereby much of the heat before it enters the area to be cooled. Whether it is desirable to do this requires careful economic study. Generally it is effective only in the case of specialized lighting application and in the construction of special equipment. When such steps are taken, it is usually necessary to intercept as much of the radiant heat as possible, usually by means of heat absorbing glasses, louvers, or baffles, which may be scrubbed by the ventilating air and re-radiated sensible heat removed. It is possible to convert and draw away as high as 80 per cent of the radiant energy from certain types of lighting installations. It is felt that lighting installations below five watts per square foot do not lend themselves economically to this type of treatment.

The illuminating engineer also has a tool through the development of new designs and new methods of applying lighting

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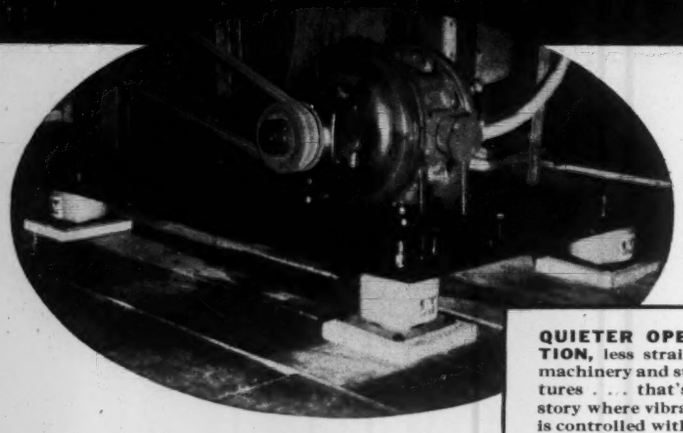

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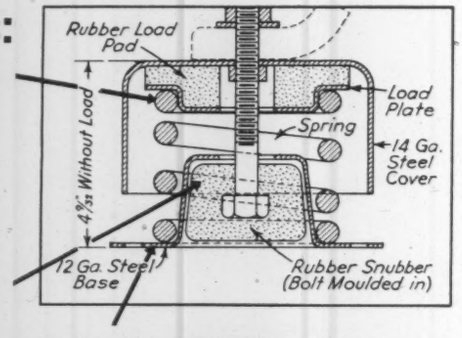

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to an interior. Large coffers and trough types of units which may be enclosed and suitably connected to duct work are often feasible. Likewise, correlation between the placing of lighting units and the location of ventilating grilles can often produce beneficial results. For example, introducing the incoming air below the stratified ceiling air often reduces the load on the cooling system by virtue of cooler recirculated air.

Railway Air Conditioning Problems

Railway air conditioning problems were discussed in a recent talk before the Chicago section of the American Society of Mechanical Engineers by A. R. Walker of the Illinois Central system. His paper is not available in published form.

Mr. Walker said that the general principles of air conditioning as applied to buildings apply also to railway passenger cars, but with the latter problems of space and weight limitations and severity of service are most important. Equipment must be designed for mounting under the car or overhead above the revenue space. In addition to keeping overall equipment dimensions to a minimum, it is imperative that weight be held to a minimum consistent with safety and reliability. To meet competition with other forms of transportation, the trend in the railway field has been toward lighter equipment and it is not consistent to lighten a car several tons at great expense and then apply heavy equipment for air conditioning.

As soon as heating and cooling load calculations are started for a car, a difference from building practice is noted. Cars are largely of metal (good conductors of heat) and the ratio of glazed surface to total wall surface is large. In motion, the cars are surrounded with moving air all the time which increases the heat transfer. Careful study of an average car shows that a refrigerating capacity between 5½ and 7 tons is needed.

The duty imposed on the evaporator unit, with from 50 to 80 passengers occupying a space of approximately 5000 cu ft, is rather severe. The required air circulation is 2000 to 2500 cfm, some of which may be recirculated, but a portion of which must be brought in from outside depending on the type of car, number of passengers, outside conditions, desired indoor conditions, odors, amount of smoking, etc. Various methods are used to distribute conditioned air to the car, a recent scheme being the use of considerable space between the ceiling and roof forming an air duct, the air being distributed to the car through a large number of small holes in the ceiling panels.

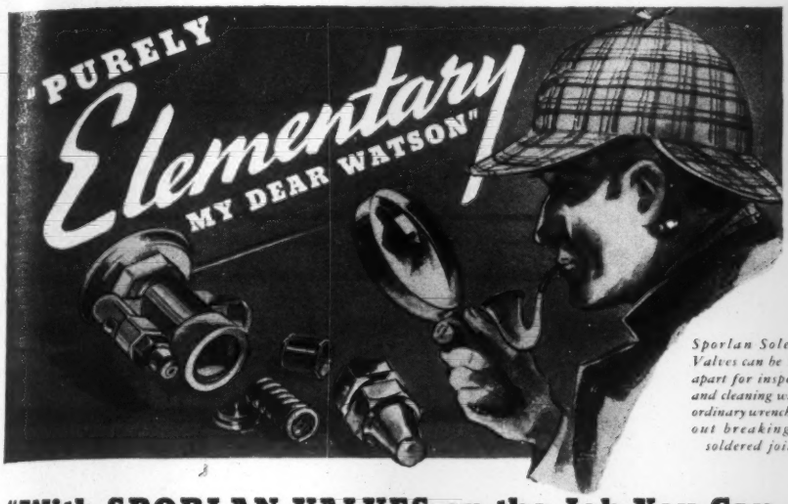
Mr. Walker gave some figures on the steam requirements for heating railway cars. He said that in severe weather an average of approximately 200 lb per hr of steam per car will be required, Pullmans needing about 250 lb, coaches 150 to 175, and baggage cars about 150 lb. Average amounts of electrical and mechanical energy, steam and ice, consumed by various railway air conditioning systems when in continuous operation at average temperatures, have been determined by tests by the division of equipment research of the Association of American Railroads:

Electro-mechanical	10.5 kw
Direct mechanical	1.0 kw and 9.8 lb
Steam ejector	3.35 kw and 230 lb steam per hr
Ice activated	1.2 kw and 463 lb ice per hr
Engine driven	0.7 kw and 7.2 lb propane per hr

Drawbar horsepower required for continuous operation of the air conditioning systems on level track at 50 mph was found to be:

Electro-mechanical	22.2 hp
Direct mechanical	19.5 hp
Steam ejector	9.3 hp and 230 lb steam per hr
Ice activated	4.5 hp and 463 lb ice per hr

In concluding, Mr. Walker said that the majority of air conditioning problems have been solved to the extent that rail-



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Sporlan Thermostatic Expansion Valves are easily taken apart for inspection, cleaning and drying...no special wrenches needed... no joints to be unsoldered.

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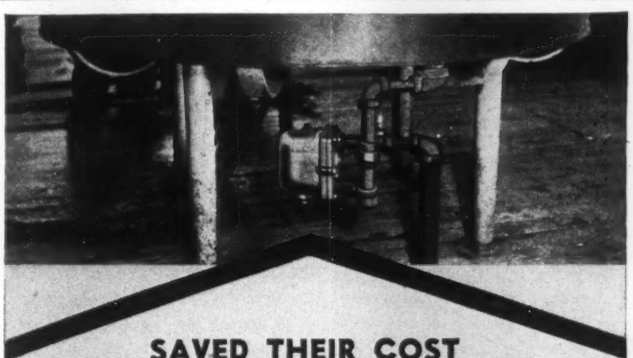

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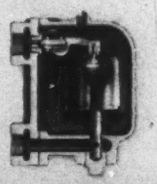

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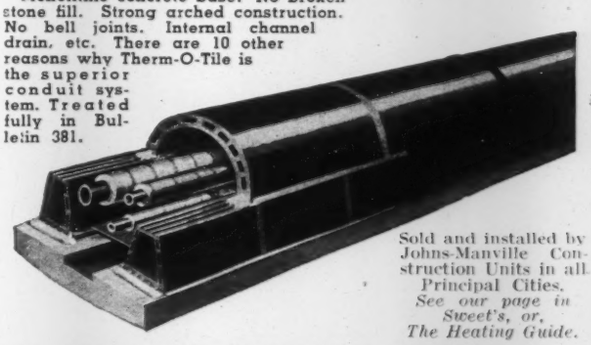

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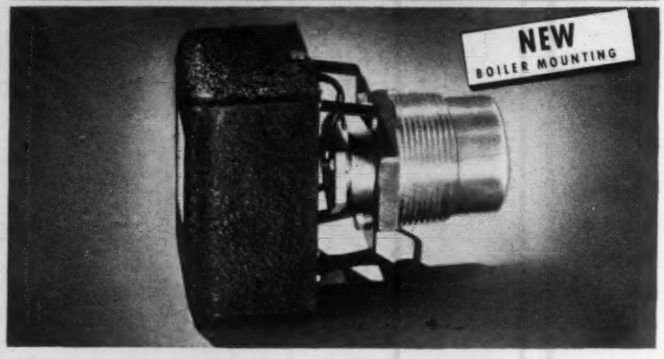

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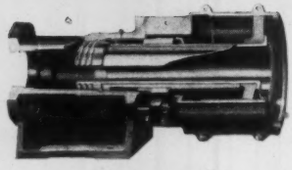

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roads are able to give reasonably satisfactory air conditioning service, although further refinement is needed and is being worked out. One of the most vexing problems is elimination of bad odors, especially in diners, club cars and smokers, where such odors cling to upholstery, curtains, etc. Keeping ducts, evaporators, and heating coils clean assists greatly, but something more is needed. Some experiments with ozone generators have proved quite satisfactory along these lines, he said, and no doubt the final answer will soon be found.

Conventions and Expositions

Air Conditioning Conference: March 28-30, Oregon State College, Corvallis, Ore. General chairman, Earl C. Willey, assistant professor of mechanical engineering. Papers include A Symposium on Fuels; Economics of Operating a Business; Condensation in Building Construction, by J. E. Lodwick; Zoning Your Heating Systems, by John Mueller; Cooling Equipment and Its Application, by Albert Freeman; Installation Codes, by C. M. MacGregor; Outlet Grilles, by R. P. Dewey; Radiator Heating, by J. F. McIndoo; Air Distribution Design, by Herbert K. Kunen; and Duct Design and Construction.

Midwest Power Conference: April 9-10, Palmer House, Chicago, Ill. Under direction of Stanton E. Winston, 401 S. Quincy St., Hinsdale, Ill. Sponsored by Armour Institute of Technology and other colleges and societies. Includes sessions on small power plants; electrical transmission; power for paper mills, refineries, and chemical plants; fuel problems; and hydro power.

International Acetylene Association: Annual convention, April 10-12, Schroeder Hotel, Milwaukee, Wis. Headquarters office, 30 E. 42nd St., New York, N. Y. Papers to be presented afternoon of April 10 include discussions of welding carbon-molybdenum pipe and welding industrial piping.

American Society of Mechanical Engineers: Spring meeting, May 1-3, Hotel Bancroft, Worcester, Mass. Headquarters office, 29 W. 39th St., New York, N. Y. Papers to be presented include Feedwater Treatment, by C. H. Fellows; Experimental Drying, by D. L. Cooper; Thermometric Time Lag, by R. Beck; Conditioned Air in Drying and Curing of Tobacco, by A. H. Cooper; Burning of Heavy Fuel Oil, by G. C. Martinson; Purchase and Use of Fuels, by E. Wadsworth Stone; Recent Developments of Oil Burners, by R. C. Vroom; Comparative Tests of Flow Measurements, by O. H. Dodkin; Thermodynamic Properties of Vapors, by Eric Leib; and The New Specific Heats, by R. C. H. Heck.

National District Heating Association: Annual meeting, May 14-17, French Lick Springs Hotel, French Lick, Ind. Secretary-treasurer, John F. Collins, Jr., 1231 Grant Bldg., Pittsburgh, Pa.

Smoke Prevention Association: Annual convention May 21-24, Hotel Statler, St. Louis, Mo. Headquarters office, City Hall Square Bldg., Chicago, Ill.

Heating, Piping and Air Conditioning Contractors National Association: Annual convention, May 27-30, Hotel Pennsylvania, New York, N. Y. Secretary, Joseph C. Fitts 1250 Sixth Ave., New York, N. Y.

National Warm Air Heating and Air Conditioning Association: Mid-year meeting, June 3-5, Palmer House, Chicago, Ill. Managing director, Allen W. Williams, 5 E. Long St., Columbus, Ohio.

Stoker Manufacturers Association: Annual meeting June 6-7, Homestead Hotel, Hot Springs, Va. Headquarters office, 307 N. Michigan Ave., Chicago, Ill.

American Society of Heating and Ventilating Engineers: Semi-annual meeting, June 17-19, Wardman Park Hotel, Washington, D. C. Secretary, A. V. Hutchinson, 51 Madison Ave., New York, N. Y.

American Society for Testing Materials: Annual meeting, June 24-28, Chalfonte-Haddon Hall, Atlantic City, N. J. Headquarters office, 260 S. Broad St., Philadelphia, Pa.

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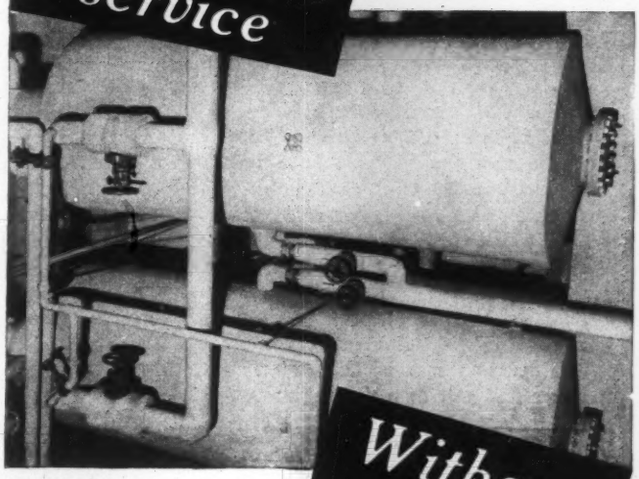

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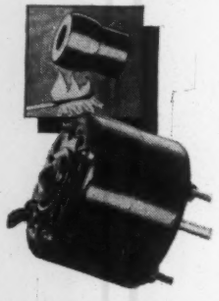

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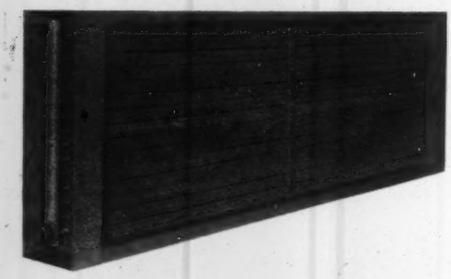

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Book Reviews

Directory of Engineering Facilities

["Directory of New England Research and Engineering Facilities." 1st ed. 1939. 118 pp., 6 x 9 in., paper cover. Published by Engineering Societies of New England, Inc., 715 Tremont Temple, Boston, Mass. Price, $2.00.]

This directory, which was published in cooperation with the New England Council by the Engineering Societies of New England, is a guide book designed to be of use to the manager or executive of a business who is in search of engineering advice and assistance. It gives in comprehensive detail the facilities, experience and industries served by some 300 private consultants and the colleges and universities in New England having laboratory and research facilities.

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Recent Trade Literature

For your convenience in obtaining copies of these bulletins, see coupon on page 150. If you write direct to the manufacturer, describe carefully what literature you want, as the number given first in each item is for use only when sending your request to HEATING, PIPING AND AIR CONDITIONING.

No. 3445. BLOWERS: W. R. Ames Co., 150 Hooper St., San Francisco, Calif. 4 p. bulletin (100) on "Utility" blowers, and 8 p. bulletin (200) on "Universal" blowers, giving approximate dimensions, performance data, capacity tables and other information.

No. 3446. BOILERS: Kewanee Boiler Corp., Kewanee, Ill. 4 p. catalog (99-2a) on Scotch marine low pressure boilers for mechanical firing for heating buildings where low ceiling heights are found and foundation excavation would not be desirable. Specifications and dimensions are included with information on construction.

No. 3447. CONDENSER TUBES: Scovill Mfg. Co., Waterville Div., Waterville, Conn. 67 p. booklet on condenser tubes giving information on corrosion, composition and physical properties of tube alloys, selection of tube material, methods of production, extrusion process, hot piercing process, cup drawing process, specifications, tube rolling methods, installation methods, and installation tools. A 6 p. engineering data section is appended.

No. 3448. CREEP STRESS DATA: Babcock & Wilcox Tube Co., Beaver Falls, Pa. Technical data card (4) giving creep stress data on "Croloys 2 and 2¼" and other materials in the same range in the high temperature tubing field. Data are arranged in chart form so that relative creep strength of each material can be seen at a glance.

No. 3449. DIESEL ENGINES: Caterpillar Tractor Co., 600 W. Washington St., Peoria, Ill. 32 p. booklet illustrating and describing 78 diesel powered machines that have operated a total of more than 1,000,000 hr, giving information on performance. Also two 2 p. leaflets giving specifications and data on new 66 kw and 52 kw diesel electric units which are self-contained and are now available in five capacities.

No. 3450. DIRECT FIRED HEATERS: Dravo Corp., Dravo Bldg., Pittsburgh, Pa. 4 p. bulletin describing the heating of hangars at the New York municipal airport with direct fired heaters.

No. 3451. DIRECTORY OF PRODUCTS: Allis-Chalmers Mfg. Co., Milwaukee, Wis. 32 p. booklet (B6057) giving a

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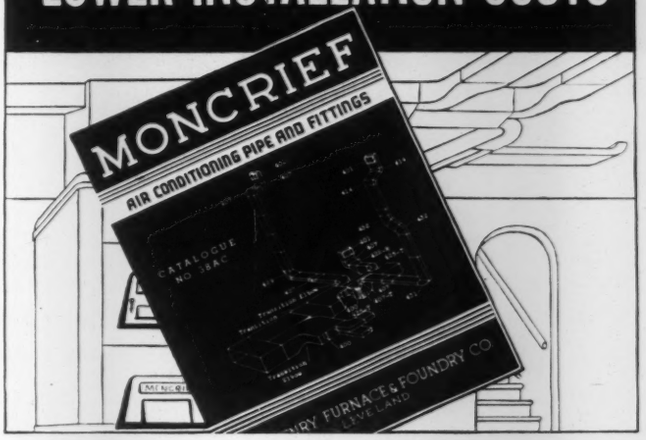

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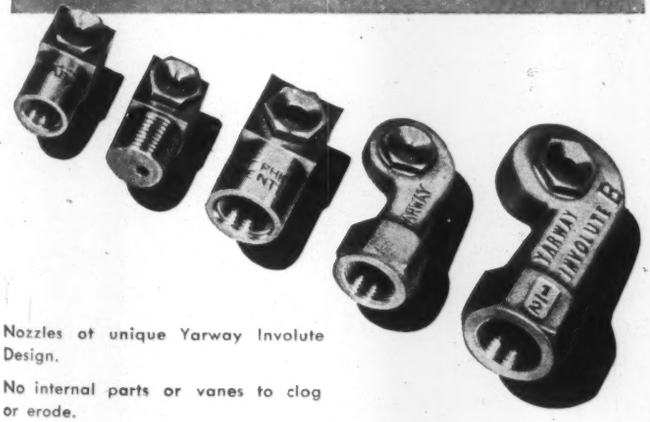

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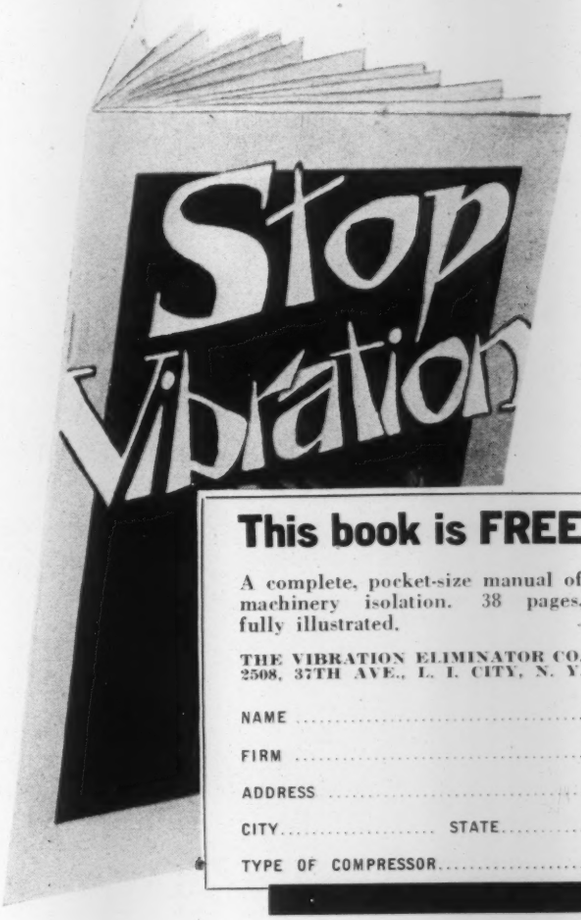

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directory of products and engineering literature of this manufacturer, including 1610 products, a chapter listing 96 different trade names, and a list of the various bulletins and other engineering literature on the equipment.

No. 3452. ELECTRIC HEATING: Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa. 38 p. illustrated catalog (28-000) on electric heating units and controls designed to meet specific industrial applications for heating liquids, solids and air, and including strip, finned, cartridge and immersion heaters, air heaters, glue and melting pots, thermostats, controls and heating elements. Six pages are devoted to application data.

No. 3453. FANS: Wagner Electric Corp., 6400 Plymouth Ave., St. Louis, Mo. 18 p. bulletin (FU-22) on air circulating fans, ventilating fans, exhaust fans, and other products, with information on applications.

No. 3454. FANS AND CIRCULATORS: Airmaster Corp., 4317 Ravenswood Ave., Chicago, Ill. 16 p. general catalog (40A) on air circulators, commercial exhaust fans, penthouses, shutters, propellers and mountings, and attic ventilating systems, illustrating and describing the equipment and giving application data.

No. 3455. GAS ENGINE COMPRESSORS: Worthington Pump & Machinery Corp., Harrison, N. J. 10 p. bulletin (L-600-B10) on angle gas engine compressors in six sizes, 375 to 1000 hp, for refineries, gas transportation lines, recycling and repressuring service, gasoline plants, refrigerating plants, gas or air lift pumping, and general air supply.

No. 3456. INSULATION: Philip Carey Co., Wayne Ave., Lockland, Cincinnati, Ohio. 16 p. booklet on "proved protection against wasted profits" suggesting a check of several factors that affect plant overhead—insulation, roofs, roof coatings, air conditioning ducts, corrugated siding, floors and similar items—and calling attention to conditions that frequently cause profit leaks and showing some of the products which cut waste.

No. 3457. LUBRICATION: Texas Co., 135 E. 42nd St., New York, N. Y. 16 p. issue of "Lubrication" (Vol. 26, No. 1) devoted to air power and compressor lubrication, with information on compressor oil characteristics, factors which control dependable operation, factors which may react against efficient operation, application of compressor oils, etc.

No. 3458. MOTORS: Electric Machinery Mfg. Co., 1331 Tyler St., N.E., Minneapolis, Minn. 2 p. data sheet (160) on high speed "packaged" synchronous motors, describing and illustrating them and giving features of construction.

No. 3459. PUMPS: Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago, Ill. 24 p. bulletin (5810D) on single stage, split case, centrifugal pumps with complete selection tables, typical specifications, principal dimensions and other engineering information to facilitate selection of such equipment. Similar bulletins are available on "Builtogether" pumps, side suction and two stage centrifugal pumps, and sewage and trash pumps.

No. 3460. ROLLER CHAIN: Morse Chain Co., Turner Pl., Ithaca, N. Y. 24 p. engineering data bulletin (R54) on roller chain for power transmission engineers, plant executives and designers, with complete information on construction, capacities and applications.

No. 3461. ROOM COOLERS: Kauffman Air Conditioning Corp., 4336 W. Pine Blvd., St. Louis, Mo. Leaflets on "packaged air conditioning" for stores, large offices and hotels; portable room coolers; window type room coolers; and room ventilators. The equipment and its features are illustrated and described.

No. 3462. RUST REMOVER: American Chemical Paint Co., Ambler, Pa. Leaflet on "Clorodine" for removing lime deposits and rust from pipe lines, boilers, condensers, engine cooling systems, water systems, etc., describing the product and its use. Also leaflet on "Deoxylyte," an inhibited acid used in connection with pickling and cleaning ferrous metals.

No. 3463. STEAM JET EJECTORS: Worthington Pump & Machinery Corp., Harrison, N. J. 6 p. bulletin (W-205-B)

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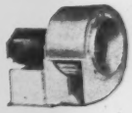

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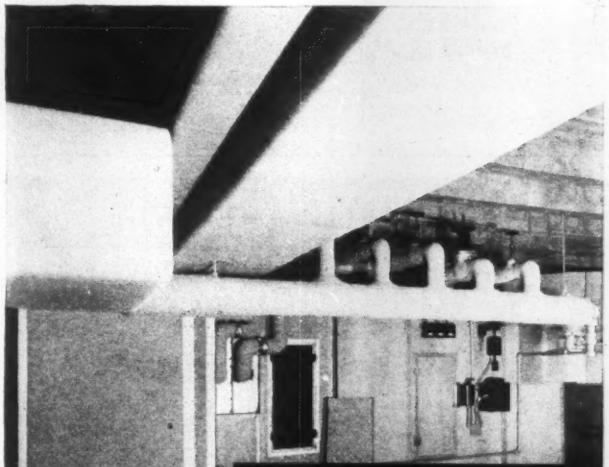

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